OPTIMUM ECONOMIC DESIGN OF PIPE/DUCT AND INSULATION THICKNESS FOR INDUSTRIAL APPLICATIONS

by

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CERTIFICATE

It is certified that the work contained in the thesi entitled " OPTIMUM ECONOMIC DESIGN OF PIPE/DUCT AND INSULATIC THICKNESS FOR INDUSTRIAL APPLICATIONS", by Sanjay Kumar Agraw has been carried out under my supervision and that this work has not been submitted elsewhere for a degree.

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NOMENCLATURE

COP	Coefficient of performance
C_{maint}	Maintenance cost, (Rs./m)
C_1	Cost of duct material for a given thickness (Rs./m^2)
C_2	Cost of insulating material (Rs./m^3)
C_3	Cost of refrigeration (Rs./ton)
C_4	Cost of pump/fan for unit volume flow rate (Rs./m^3/min)
C_5	Cost of coal (Rs./kg)
d_1	Inner diameter of pipe/duct (m)
d_2	Outer diameter of pipe/duct (m)
d_e	Equivalent diameter of duct (m)
e	Pipe roughness (m)
F_1	Percentage installation factor
F_2	Factor of safety
F_3	Percentage friction loss factor
F_4	Boiler maintenance factor
F	Frictional loss
f	Frictional factor
Gr	Grashof number

H_1	Head of the pump (m)
H_2	Head of the fan (m)
h_o	Outer heat transfer coefficient ($kJ/m^2.h.C$)
h_i	Inner heat transfer coefficient ($kJ/m^2.h.C$)
K_I	Insulation conductivity (kJ/m.h.C)
K_f	Fluid conductivity (kJ/m.h.C)
L	Life of the system (years)
Nu	Nusselt number
OF	Operating factor
P_{rf}	Prandtl number
ΔP	Pressure loss (N/m^2)
P_{wT}	Theoretical power for water flow (kW)
P_{aT}	Theoretical power for air flow (kW)
$\dot{Q}$	Heat loss/gain across the pipe/duct boundary (kJ/h)
r	Interest rate
R_{ed}	Reynolds number
SPWF	Series present worth factor
T_i	Fluid temperature (K)
T_1	Inner wall temperature (K)
T_s	Surface temperature (K)
T_a	Ambient air temperature (K)
u	Fluid velocity (m/s)

V	Volume flow rate (m^3/min)
V_a	Wind velocity (m/s)
ν_f	Kinematic viscosity (m^2/s)
σ	Stefan Boltzman constant (kJ/h.K^4)
ε	Emissivity of surface covering
ρ	Density (kg/m^3)
η_p	Pump efficiency
η_m	Motor efficiency
η_f	Fan efficiency
η_b	Boiler efficiency

ABSTRACT

A general computer program has been developed for the insulated pipe and duct systems to get optimum insulation thicknes and pipe/duct diameter based on total cost comprising initial and running expenditures over the entire life. Effects of the velocit emissivity, radiation loss/gain and heat generated due to frictior of fluid and inefficiencies in the system have been considered to make the results rather more realistic. Results have been present in the tabular form for various fluid and ambient temperatures and capacities.

Analyses were carried out for utilization of optimum dimensions for different applications. Sensitivity analysis shows the effect of design variables on the total cost, refrigeration loss and electrical energy . It has been found that 25%, 16% and 18% electrical energy could be saved by selecting the dimensions higher than the optimum values by incurring 3.5% additional total cost over the optimum total cost for the insulated duct and pipe systems and heat loss through insulated pipe, respectively . To have better understanding, the results for this are presented in graphical form.

CHAPTER-1

INTRODUCTION

1.1 NEED FOR ENERGY CONSERVATION

The ultimate aim of any engineer engaged in the design of thermal systems it to produce a financially feasible and economical viable design which achieves a desired performance level. In the past, when energy cost were low and and its availability potential was high, the energy required for the functioning of any system or the running cost over its life time was a matter of secondary importance. But now with limited energy resources and high cost, i is imperative to conserve energy and use it optimally. A system design should comply with these concept and render service without affecting its performance.

Energy consumption has assumed paramount importance in the recent times. This is mainly due to energy crisis resulting from depleting reserve of fossil fuels. But to meet the ever increasin demand of energy has become rather serious problem. As such attem are made to find out various alternatives either to use different systems which do not need electrical energy or will need as less a necessary as compared to the older inefficient systems which were nothing but " Energy Hogs".

For such systems which run round the clock for the whole ye the energy consumption becomes even more important consideration. Pipe and ducts which are required in refrigeration and airconditio thermal system, process industry etc. generally remain in operatio for almost round the year or so. The total cost of any system therefore depends not only on the initial cost or the fixed cost b also on the running cost. The running cost depends on the power consumption. Generally, systems with high initial costs (Involvin sophisticated design concepts) are presumed to have lower running costs with high levels of performance, but such systems may not be always economically feasible. The decision therefore to be taken the type of insulant, surface covering and system dimensions is m on the economic consideration.

The total cost of pipe or duct installation consists not on of the initial cost and the running cost but also of the maintenan cost and the requirement of spares and consumables. One has there to make a complete analysis of the total costs involved on the ent life span. This depends upon the type of system and taken on the basis of standard practice. Hence the final decision is made in accepting one set of dimensions or the other.

1.2 LITERATURE REVIEW

In general, attention has been focussed either on economic insulation thickness (for a given pipe diameter) or on the optimum pipe diameter (without insulation) rather than finding both the dimensions simultaneously for a given mass flow rate.

Many short cut methods and simplified equations are also available for estimating optimum pipe diameter [1-7]. Some gene "Thumb Rules" used to get pipe diameters are given in table 1.1.

Table 1,1 Thumb Rules for economic velocities for sizing steel pipe lines having turbulent flow [1].

Type of fluid	Reasonable velocity (m/s)
Water or fluid similar to water	1 - 3
Low pressure steam (1.723 bar gauge)	15 - 30
High pressure steam (6.891 bar gauge)	30 - 60
Air at ordinary pressure (1.723 to 3.446 bar gauge)	15 - 30

Various nomographs and charts are also available [1-4] for estimating the optimum diameter of pipe and duct under ordinary conditions. When unusual conditions are involved or when a more accurate determination of the optimum diameter is desired, followi equation is available for turbulent flow in steel pipes [1]. But it does not include pump cost and refrigeration loss cost.

$$d_{1,opt} = q_f^{0.448} \rho^{0.132} \mu^{0.025} \left[\frac{0.88\ K\ (1+J)\ H_Y}{(1+F)\ XE\ K_f} \right]^{0.158}$$

Where,

q_f = Volume flow rate (ft^3/s)

ρ = Fluid density (lb/ft^2)

μ = Fluid viscosity (centipoises)

K = Cost of electrical energy ($/kWh)

H_y = Hours of operation per year

E = Efficiency of motor and pump expressed as a fraction

F = Ratio of total costs for fittings and installation to purchase cost for new pipe

K_F = Annual fixed charges including maintenance, expressed as a fraction of initial cost for completely installed pipe

X = Purchase cost of new pipe per foot of pipe length if pipe diameter is 1 in ($/ft)

J = Frictional loss due to fittings and bends, expressed as equivalent fractional loss in a straight pipe.

A microcomputer program to determine the heat loss through insulated piping has been written by Stephen [8] in BASIC. However, the surface temperature of the pipe is assumed to be the same as the fluid temperature in the pipe.

The computation of economical insulation thickness for cylindrical bodies has been done after ignoring the dependence of convective heat transfer coefficient on temperature and diameter of pipe. Nevertheless the thermal resistance does change with varying

thickness of insulation [9]. Therefore the simple expression fc economical insulation thickness is not very correct.

Economical insulation thickness has been determined by Zahn [10] for flat surface using series present worth method for computation of the insulation cost per year. Arent, et al [11] have calculated above thickness with the help of the McMillan's method [12]. While the mathematical model of stone [13] employs the simple interest. However, the temperature, and diameter dependence of convective heat transfer has not been considered.

Prasad [14] has determined the economical insulation thickness by considering the temperature and diameter dependence of convective heat transfer. But it cannot, unfortunately, be applied to the cases where heat loss due to radiation is significant as it does not include radiation losses. Gupta and Prasad [15] have developed a mathematical model for the design of a duct system for optimal energy conservation using present worth method, but it does also not consider the effect of wind velocity and radiation losses the optimal dimensions.

With limited availability of useful energy and consequent us of costs, the designers are made think for the optimal use of energ Serious steps are being taken in this direction through ASHRAE standard 90-75 on energy conservation [16]. Some work has been initiated in this direction even earlier. Bonar [17] has analysed the different parameters and factors that affect the economics of a refrigeration system for a warehouse with reference to the minimization of energy costs.

Stephen [18] has discussed about the economic insulation thickness while Spielvogel [19] has found that more insulation can increase the energy consumption. Joseph [20] and Ross [21] have discussed the recent trends and energy saving procedure in the design of refrigerated warehouses.

1.3 PRESENT WORK

A general computer program has been developed to optimize a function of n variables. Optimization has been done for the following cases:

i. Water flowing through the pipe,

ii. Air flowing through the duct,

iii. Superheated steam flowing through the insulated pipe ar

iv. Sensitivity analysis for the above three cases has bee done in order to ascertain rather more practical data.

In the present work optimum dimensions have been obtained with respect to the minimum total cost taking into consideration the effects of radiation, wind velocity, surface emissivity of the insulation covering and heat generated due to fluid friction and pump/fan in efficiency.

Optimum dimensions have been obtained for various combinatio of fluid and ambient temperatures at various mass flow rate. The study reveals that 16% electricity for the pipe system, 25% electri-city for the duct system and 18% electricity for the heat loss

through insulated pipe could be saved at the cost of 3.5% additional total cost compared to the optimum. Optimum velocities for the pipe system has been found to lie between 1.15 to 1.53 m/s for volume flo rate between 1 to 25 m^3/min of water and ambient temperature between 17 C to 47 C. Whereas its value for the duct system lies between 8.0 to 10.9 m/s for volume flow rate between 10 to 2000 m^3/min and ambient temperature between 32 C to 47 C.

Sensitivity analysis has been carried out to show the effects of design variables on the total cost, electrical energy and refrigeration loss. The electrical energy and total are more sensitive to the inner diameter of the system as compared to insulat thickness. Thus for a known set of design parameters (viz volume flow rate, fluid temperature, insulation conductivity, operating factor etc.) and for a given design ambient temperature and wind velocity of a given location, an optimum system can be recommended which include the size of pump, size of refrigeration plant, pipe diameter and insulation thickness.

CHAPTER-2

PROBLEM FORMULATION

2.1 MATHEMATICAL MODELLING

Figure 2.1 shows an insulated pipe/duct meant for transporting chilled water/conditioned air in refrigeration systems or transporting hot water in the process heating or supplying super heated steam in thermal power plant. Water/air is cooled or heate up to the desired temperature in the refrigeration or heating. Fro the outlet of the plant it is transported through the insulated pip duct to the chamber(s) as per required applications.

A pipe (duct) system can be viewed as a thermodynamic system wherein fresh water (air) or recirculated water (air) enters at one end and leaves it at another end after being chilled (conditioned) in the system. Energy interaction in the form of work takes place across the pipe (duct) boundary. Main components of such a pipe (duct) system are:

i. Pipe or duct
ii. Insulating material
iii. Chilling/heating or refrigeration system and
iv. Pump with motor or Blower/fan

The system is considered to be the best one when it does not allow any refrigeration (heating) loss and requires the least total cost consisting of initial and running costs.

The exact diameter of the pipe (duct) and thickness of insulating material is estimated by an economical analysis i.e., the minimum total cost.

Among many factors, insulation and pumping costs can be taken as the prominent factors that greatly influenced not only the initial cost of the system, but its operating cost also. Therefore diameter of the pipe (duct) and thickness of insulation is found in such a way so that there is a balance point between economy and energy conservation. The insulation thickness corresponding to the minimum total cost is termed as the "Economic insulation thickness" and pipe (duct) diameter as the "Economic pipe diameter". The values of the Economic pipe diameter and insulation thickness are determined by combining the principles of the fluid dynamics with cost considerations.

Following assumptions are made in order to formulate the mathematical model of the problem.

i. Pipe/duct material thickness is not considered as its value is extremely small in comparison to pipe/duct dimensions and insulation thickness.

ii. Temperature of inner and outer water films or air films are taken as those of the inner and outer surface thereferatures respectively.

iii. Fouling factors on both sides of the pipe/duct are taken to be insiginficant compared to other thermal resistances.

iv. Scrap value of the pipe/duct after its complete life span, and the cost of the floor space are not considered in view of larger life being considered.

v. Cost involved in sound attenuation is not taken in to account.

vi. The analysis is carried out for steady state case.

2.1.1 ESTIMATING HEAT LOSS/GAIN AND SURFACE TEMPERATURE

Figure 2.1 exhibits an insulated pipe/duct having surface temperature T_s, kept in surrounding at temperature T_a. Then, for an insulated pipe/duct, steady state heat transfer equation across the pipe/duct wall per unit length of pipe/duct can be written as:

$$\dot{Q} = \pi \, d_2 \left[h_o (T_a - T_s) + \sigma \varepsilon (T_a^4 - T_s^4) \right] \tag{2.1a}$$

$$= \frac{2\pi K_I}{\ln(d_2/d_1)} \left[T_s - T_I \right] \tag{2.1b}$$

$$= \pi d_1 h_i (T_I - T_i) \tag{2.1c}$$

In Eq. (1.1a) first and second terms represent the convective and radiative heat losses, respectively. Inner and outer film heat transfer coefficients (h_i and h_o) are taken from the well known equations available in the literature [22].

For turbulent flow of fluid inside the pipe/duct we have,

$$h_i = \frac{Nu \, K_f}{d_1} \tag{2.2}$$

Where,

$$Nu = 0.023\, R_{ed}^{0.8}\, (P_{rf})^{n} \tag{2.3}$$

$$n = 0.4 \text{ for heating}$$

$$\quad\;\; 0.3 \text{ for cooling}$$

From Eq. (2.2) and Eq. (2.3) we get,

$$h_i\, d_i = 0.023\, R_{ed}^{0.8}\, (P_{rf})^{n}\, K_f$$

$$= 1.05\text{x}10^{-3}\, K_f\, \left(\frac{V}{\nu_f\, d_1}\right)^{0.8} (P_{rf})^{n} \tag{2.4}$$

Free convective heat transfer coefficient with D values as in table 2.1 is given by [23]:

$$h_o = D\left(\frac{T_a - T_s}{d_2}\right)^{0.25} \tag{2.5}$$

Equation (2.5) is true only for still air. To account for the effect of air velocity on heat transfer from surface to air, Langmuir shows [12] that convection is increased by air circulation, according to the equation,

$$h_o = D\left(\frac{T_a - T_s}{d_2}\right)^{0.25} (1+ 2.86\, V_a)^{0.5} \tag{2.6}$$

From Eq.(2.1b) and Eq.(2.1c), alter simplification we get,

$$T_s = T_i + \frac{\dot{Q}}{\pi}\left[\frac{\ln\,(d_2/d_1)}{2K_I} + \frac{1}{d_1\, h_i}\right] \tag{2.7}$$

Substituting for $\dot{Q}$ from Eq. (2.1a) in Eq. (2.7) we get,

$$T_s = T_i + d_2 \left[\frac{\ln(d_2/d_1)}{2K_I}\right] \left[h_o(T_a - T_s) + \sigma\varepsilon(T_a^4 - T_s^4)\right] \quad (2.8)$$

Equation (2.8) being a non-Linear in T_s has been solved numericall for T_s and then $\dot{Q}$ is obtained form Eq. (2.1a) as:

$$\dot{Q} = \pi\, d_2 \left[\, h_o\ (T_a - T_s) + \sigma\,\varepsilon\,(T_a^4 - T_s^4)\right]$$

2.1.2 ESTIMATION OF POWER REQUIREMENT

2.1.2.1 POWER FOR PUMP

For pumping fluid power is supplied by the pump/fan to overcome frictional resistance, change in elevation, change in internal energy, and other pressure losses depending upon the system net-work. In any real flow, frictional losses constitute the main component for which the fanning equation is used [1].

$$dF = -\frac{dp}{\rho} = 2C_f\,\frac{u^2}{d_1}\,dl \quad (2.9)$$

Where C_f = Fanning friction factor

The fanning friction factor (C_f) is related to the friction factor (f) given by moody as:

$$C_f = f/4 \quad (2.10)$$

By combining Eqs. (2-9 and 10), we get:

$$dF = -\frac{dp}{\rho} = \frac{f\, u^2\, dl}{2d_1} \tag{2.11}$$

The friction factor f is based on experimental data and is a function of Reynolds number and the relative roughness of the pipe (e/d_i). In the turbulent flow region, the relative roughness of the pipe has a large effect on the friction factor. The values of the equivalent pipe roughness are given in table 2.2 These values are only approximations, even for new pipe, and the values may increase because of surface pitting and corrosion after the pipe is in service.

Approximate equation (upto $\pm$ 5% accuracy) showing the relationship between the friction factor and the Reynolds number in the turbulent flow region is [2]:

$$f = 0.0055 \left[1 + \left(20000 \frac{e}{d_1} + \frac{4.7124\text{x}10^7 \nu_f\, d_1}{V}\right)^{1/3} \right] \tag{2.12}$$

Since velocity, density and viscosity of the flowing fluid remain constant and the pipe diameter is uniform over a total pipe length (l), Eq. (2.11) can be integrated to get the frictional loss per unit length of pipe as:

$$\frac{F}{l} = \frac{-\Delta p}{\rho l} = \frac{fu^2}{2d_1} \tag{2.13}$$

Therefore pressure drop per unit length of pipe is,

$$\Delta p_w = \rho f \frac{u^2}{2d_1}$$

$$= 1.2384\text{x}10^{-6} \frac{\rho V^2}{d_1^5}\left[1 + \left(20000 \frac{e}{d_1} + \frac{4.7124\text{x}10^7 \nu_f d_1}{V}\right)^{1/3}\right] \tag{2.14}$$

Theoretical Pump Power required to overcome frictional loss per unit length of pipe would be:

$$P_{wT} = \frac{\Delta p\ V}{60000}$$

$$= 2.063\text{x}10^{-11} \frac{\rho V^3}{d_1^5}\left[1 + \left(20000 \frac{e}{d_1} + \frac{4.7124\text{x}10^7 \nu_f d_1}{V}\right)^{1/3}\right] \tag{2.15}$$

In a strict sense, Eq. (2.14) is limited to conditions in which flowing fluid is incompressible and temperature of fluid is constant. When dealing with compressible fluid such as air, steam its good engineering practice to use Eq. (2.14) only if the pressure drop over the system is less than 10% of the initial pressure. In our case both these conditions are satisfied, so Eq. (2.14) is used to calculate the frictional pressure drop.

2.1.2.2 POWER FOR FAN

Pressure drop per unit length of duct for air flow is expressed by [24]:

$$\Delta p_a = 9.825\ \text{x}10^{-6} \frac{V^{1.845}}{d_1^{4.92}} \tag{2.16}$$

Theoretical Fan Power to circulate Vm^3/min of air would be:

$$P_{aT} = 1.6366 \times 10^{-10} \frac{V^{2.845}}{d_1^{4.92}} \quad (2.17)$$

2.1.3 PRESENT-WORTH METHOD

In this method all the costs and incomes are translated into the present worth. In order to determine the present worth of a future sum, first the interest rate is established. Then present worth is the value of a sum of money at the present time that, with compound interest, will have a specified value at a certain time in future. Thus

$$p = \frac{S}{(1+r)^m} = (PWF)S \quad (2.18)$$

Where,

p = principal

r = rate of interest per period

S = total amount to be repayed at future time

m = number of periods and

PWF = single-payment present worth factor

The series present worth is given by

$$p = R\left[\frac{(1+r)^m - 1}{r(1+r)^{m-1}}\right] = R\ (SPWF) \quad (2.19)$$

Where p = Original amount

R = amount withdrawn at the end of first and subsequen periods and

SPWF = series present worth factor

2.1.4 SENSITIVITY ANALYSIS

In practice, a designer would be interested in knowking how the total cost, electricity consumption or refrigeration loss varies with a change in the design parameters in the vicinity of the optimum value. This type of sensitivity analysis helps the designer compute the changes and ascertain the choice of more realistic values.

Further in some cases, the results obtained from the optimization procedure may have to be rounded-off to the nearest practical values of the design variables. Hence a sensitivity analysis of the refrigeration loss, electricity consumption and total cost with respect to different design variables is conducted at the optimum point. Thus the reference design is taken as the optimal design, and the values of the design parameters are change by 50% in step of 10% on the positive and negative sides.

2.2 OBJECTIVE FUNCTIONS

2.2.1 FOR PIPE SYSTEM

In the present work the objective function considered is the total cost per unit length of pipe for the entire life span in terms of the present value. The total cost of the pipe system is the sum of its initial cost and running cost. Initial cost comprise the costs of pipe,insulation, refrigeration plant, water circulati pump and their installation costs. While running comprises of the costs of electric power required to run the pump, refrigeration lo occuring across the pipe boundary and maintenance. Formulation ha

been done for the unit length of pipe.

A plot of the logarithm of thepipe diameters versus the logarithm of the purchase cost per meter of pipe is essentially a straight line (In the range of 100 to 500 mm). Therefore, the purchase cost of pipe (C_{pipe}) is represented by:

$$C_{pipe} = 7500\ d_1^{1.5} \quad (\text{For } 0.10 \leqslant d_1 \leqslant 0.50) \tag{2.20}$$

Purchase cost of insulation material ($C_{insulation}$) is :

$$C_{insulation} = \frac{\pi}{4} (d_2^2 - d_1^2)\ C_2 \tag{2.21}$$

Therefore actual cost of piping system (pipe + insulation) is expressed as follows:

$$C_{piping} = [C_{pipe} + C_{insulation}] (1 + F_1) \tag{2.22}$$

Fixed charges for the pump and motor (C_{pump}) from Eq.(2.14) is:

$$C_{pump} = C_4\ V\ (\frac{\text{head loss in pipe}}{\text{original head of pump}})\ (1+F_2)$$

$$= \frac{C_4\ V\ \Delta p_w\ (1+F_3)\ (1+F_2)}{9800\ H_1} \tag{2.23}$$

In the formulation of Eq. (2.23), we have made the assumpti that for a given capacity, cost of pump increases linearly with hea of the pump.

Fixed charges for the refrigeration system ($C_{ref^n\ system}$) from Eq. (2.1a and 2.15) is :

$$C_{ref^n system} = \frac{1}{3.5x3600} C_3 \ (\dot{Q} + 3600 \frac{P_{wT}}{\eta_p} \ (1+F_3)) \ (1+F_2)$$

$$= 7.9365x10^5 C_3 \ (\dot{Q} + 3600 \frac{P_{wT}}{\eta_p} \ (1+F_3)) \ (1+F_2) \qquad -(2.24$$

Where second term is the bracket $3600 \frac{P_{wT}}{\eta_p} (1+F_3)$ represents the frictional heat dissipation due to the flow of water.

Therefore,

$$C_{initial} = [C_{piping} + C_{pump} + C_{ref^n system}] \qquad (2.25$$

For the cost of electricity (a logistic curve)

$C_{eJ} = \frac{1.293}{0.97886 + e^{-0.09338 \ (J+7)}}$ has been taken to forecast the future costs of electricity [25]. Thus the cost rates of electricity (Rs/kWh) C_{e1}, C_{e2}, -----C_{eL} in the Ist, 2nd, --- Lth year is obtained. The net present value of these costs can be computed as:

$$C_{e, \ effective} = \sum_{J=1}^{L} \frac{C_{eJ}}{(1+r)^{J-1}} \qquad (2.26$$

So, cost of electric power for the pump (C_{pp}) from Eq. (21.5 & 2.26 for the whole life is :

$$C_{PP} = 8760 \ OF \frac{P_{wT}}{\eta_p \ \eta_m} C_{e,effective} \ (1+F_3) \qquad (2.27$$

Where OF is the part of the day for which system has to operate and is called 'operating factor'.

The cost of refrigeration loss ($C_{ref^n loss}$) for the whole life is:

$$C_{ref^n}\ loss = \frac{8760}{COPx3600}\ OF\ (\dot{Q} + 3600\ \frac{P_{wT}}{\eta_p}\ (1+F_3))\ C_{e,effective} \tag{2.28}$$

Using COP = 3, Eq. (2.28) renders

$$C_{ref^n}\ loss = 0.811\ (\dot{Q}+3600\ \frac{P_{wT}}{\eta_p}\ (1+F_3))\ OF\ C_{e,effective} \tag{2.29}$$

Maintenance cost has been considered as 10% of the equipment cost [26] plus insulation cost.

$$C_{maint} = 0.10[C_{insulation}\ (1+F_1) + C_{pump} + C_{ref^n system}$$

$$\left[\frac{(1+r)^{L} - 1}{r(1+r)^{L-1}} \right] \tag{2.30}$$

Therefore,

$$C_{running} = [\ C_{ref^n}\ loss + C_{pp} + C_{maint}] \tag{2.31}$$

The total cost (C_T) is then given by:

$$C_T = C_{initial} + C_{running} \tag{2.32}$$

Its functional dependence is expressed as:

$$C_T = C_T\ (T_a,\ T_i,\ V,\ d_1,\ d_2,\ \dot{Q},\ \nu_f,\ e,\ \varepsilon\ ,\ C_2,\ C_3,\ C_4,\ K_f,\ K_I,\ OF\ ) \tag{2.33}$$

Our aim is to minimize C_T subject to the constraints

$$d_1 \geqslant 0 \tag{2.34}$$

$$d_2 \geqslant 1.02\ d_1 \tag{2.35}$$

These relations express the fact that inner diameter of pipe should be greater than zero if there is any flow to exist and insulation thickness ($\frac{d_2 - d_1}{2}$) should be greater than zero for it to have any meaningful value.

2.2.2 FOR DUCT SYSTEM

It is the most common practice to use rectangular ducts (sides a & b) for transportating conditioned air from the central duct to various buildings scattered nearby. It can be replaced by an equivalent circular duct having the same flow rate and equal pressure drop in the same length as [27]:

$$d_e = 1.302 \left[\frac{(a.b)^5}{(a+b)^2} \right]^{0.125} \tag{2.36}$$

The formulation for duct system has been done for the case of circular ducts as frictional pressure drop is available for such canfiguration. Then for a given aspect ratio of duct (a/b), we can get the dimensions of rectangular duct using Eq. (2.36). Unit length of duct has been considered in the formulation.

Duct cost (C_d) is :

$$C_d = \pi \, d_e \, C_1 \, (1 + F_1) \tag{2.37}$$

The cost of sheet materials for airduct is given in Appendix A.

Insulation cost (C_i) is :

$$C_i = \frac{\pi}{4} (d_2^2 - d_e^2) \, C_2 \, (1 + F_1) \tag{2.38}$$

Refrigeration system cost (C_r) from Eqs. (2.1a & 2.17) is:

$$C_r = \frac{1}{3.5x3600} C_3 \left(\dot{Q} + 3600 \frac{P_{aT}}{\eta_F} (1 + F_3)\right)(1+F_2)$$

$$= 7.9365x10^{-5} C_3 \left(\dot{Q} + 3600 \frac{P_{aT}}{\eta_f} (1+F_3)\right)(1+F_2) \quad (2.39)$$

Fan cost (C_f) from Eq. (2.16) is:

$$C_f = C_4 V \frac{\Delta pa (1+F_3) (1+F_2)}{9800 H_2} \quad (2.40)$$

Therefore,

$$C_{initial} = [C_d + C_i + C_r + C_f] \quad (2.41)$$

Cost of electric power for the fan (C_p) from Eq. (2.17) for the whole life is :

$$C_p = 8760 \text{ OF } \frac{P_{aT}}{\eta_f} C_{e,effective} (1+F_3) \quad (2.42)$$

Cost of refrigeration loss (C_q) for the whole life is :

$$C_q = 0.811 \left(\dot{Q} + 3600 \frac{P_{aT}}{\eta_f} (1+F_3) \text{ OF } C_{e,effective}\right. \quad (2.43)$$

$$C_{maint} = 0.10 [C_i + C_r C_f] \left[\frac{(1+r)^L - 1}{r(1+r)^{L-1}}\right] \quad (2.44)$$

Therefore,

$$C_{running} = [C_p + C_q + C_{maint}] \quad (2.45)$$

The total cost (C_T) is then given by :

$$C_T = C_{initial} + C_{running} \quad (2.46)$$

Our aim is to minimize the objective function, C_T, subject to constraints:

$$d_1 \geqslant 0 \qquad (2.47)$$

$$d_2 - d_1 \geqslant 0 \qquad (2.48)$$

2.2.3 FOR HEAT LOSS FROM INSULATED PIPE

The cost of pipe, insulation and piping are the same as for the pipe system and are given by Eq. (2.20, 2.21 and 2.22), respectively.

Running cost of boiler (C_b) per unit length of pipe from Eq. (2.15) for the whole life is:

$$C_b = 8760x\ OF \frac{(\dot{Q} - 3600\ P_{wT}\ (1+F_3))}{\eta_b\ (\Delta H_f)} C_6 \left[\frac{(1+r)^L - 1}{r(1+r)^{L-1}} \right] \qquad (2.49)$$

Values of K_f, ν_f and for steam are given in Appendix B at different temperatures.

The maintenance cost (C_{maint}) of the system is:

$$C_{maint} = 0.10x\ C_{insulation}(1+F_1)\ (1+F_4) \left[\frac{(1+r)^L - 1}{r(1+r)^{L-1}}\right] \qquad (2.50)$$

The total cost (C_T) is then given by:

$$C_T = C_{piping} + C_b + C_{maint} \qquad (2.51)$$

Our aim is to minimize this C_T subject to constraints given by Eqs. (2.34 & 2.35).

Table 2.1

D Values for water or air [23]

Configuration	D Values $10^4 < Gr\ Pr_L < 10^9$
Horizontal Cylinder	4.647
Vertical plates or cylinder	4.992

Table 2.2

Values for e for various pipe construction materials

Material of construction	Equivalent roughness for new pipe e (meter)
Drawn tubing	0.0000015
Commercial steel	0.000045
Wrought Iron	0.000045
Galvanized Iron	0.000150
Cast Iron	0.000255
Concrete	0.0003 to 0.003

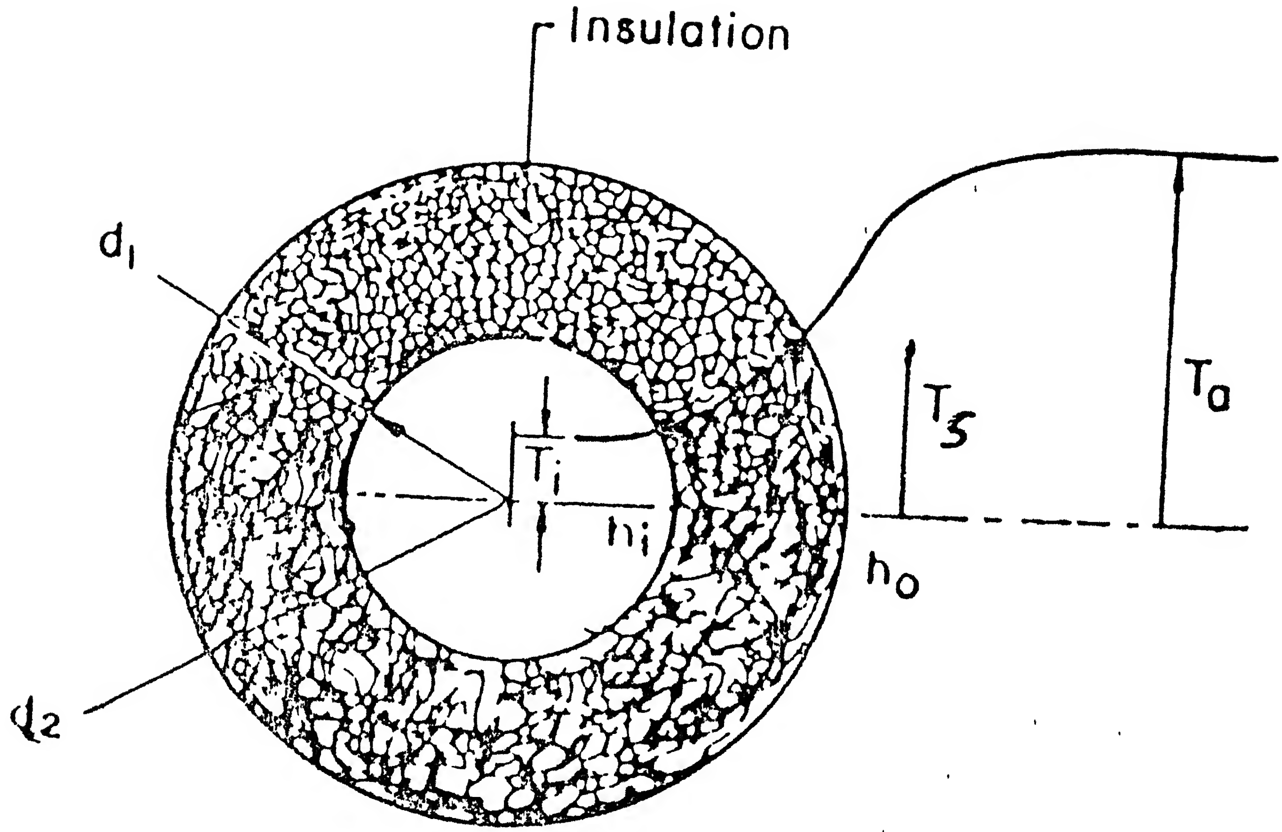

Fig.2·1 An insulated circular duct/pipe

CHAPTER-3

OPTIMIZATION

3.1 OPTIMIZATION TECHNIQUE

The presence of constraints in a nonlinear programming problem creates more problems while finding the minimum. Several situations can be identified depending on the effect of constraints on the objective function. The simplest situation is when the constraints do not have any influence on the minimum point. From the application point of view on the real problems, there is a possibility of having two or more local minima entirely due to the nature of the objective function contours. By taking into account of all these possibilities, in the present analysis, Gradient Projection method of non-linear programming constraint optimization technique builtin the HP 9000 system named as EO4UCF is used.

3.2 BACKGROUND TO OPTIMIZATION METHOD

Algorithm of the optimization program E04 UCF generate an iterative sequence $\{x^{(K)}\}$ that converges to the solution X* in the limit. To terminate computation of the sequence, a convergence test is performed to determine whether the current estimate of the soluti[on] is an adequate approximation. This method construct a sequence $\{X^{(K)}\}$satisfying:

$$X^{(K+1)} = X^{(K)} + \alpha^{(K)} P^{(K)}$$

Where the vector $P^{(K)}$ is the direction of search, and $\alpha^{(K)}$ is the steplength. The steplength $\alpha^{(K)}$ is choosen so that $F(X^{K+1}) < F(X^{(K)})$ and is computed using the technique of one dimensional optimization which requires function evaluation only and fits a quadratic polynominal.

Bounds on the variables are dealt with by fixing the variables on their bounds to minimize the function. By examining estimates of the lagrange multipliers it is possible to adjust the set of variables fixed on their bounds so that eventually the bounds active at the solution should be correctly identified. This type of method is called an active set method. One feature of such method is that, given an initial feasible point, all approximations $X^{(K)}$ are feasible. This approach is extended to general linear constrain At a point, X, the set of constraints which hold as equalities being used to predict, or approximate, the set of active constraints is called the working set.

Non-linear constraints are more difficult to handle. The method currently implemented in the program handle non-linearly constraint problem by transforming it into a sequence of quadratic programming problem.

3.3 DESCRIPTION OF THE PROGRAM

The program E04UCF is designed to minimize an arbitrary smoot function subject to constraints, which include simple bounds on the variables, linear constraints and smooth non-linear constraints.

The user provides the subroutines that define the objective and constraint functions and as many of their first partial derivatives as possible. Unspecified derivatives are approximated by finite differences. This program uses a sequential quadratic programming (SQP) algorithm in which the search direction is the solution of a quadratic programming (QP) problem. The algorithm treats bounds, linear constraints and non-linear constraints separately.

The problem is assumed to be stated in the following mathematical form:

Minimize F(X)

$$\text{Subject to: } l \leqslant \begin{bmatrix} X \\ A_L X \\ C(X) \end{bmatrix} \leqslant u, \qquad (3.1)$$

Where F(X), the objective function, is a non-linear function, A_L is an n_L by n constant matrix, and C(X) is an n_N element vector of non-linear constraint confunctions. The objective function and the constraint functions are smooth, i.e, at least twice-continuousl differentiable. In this program upper and lower bounds are specifie for all the variables and for all the constraints. An equality constraint is specified by setting $l_i = u_i$.

The objective function is defined by subroutine OBJFUN, and the non-linear constraints are defined by sub-routine CONFUN. On every call, these subroutines return appropriate values of the objective and the non-linear constraints in OBJF and C. Just befor either OBJFUN or CONFUN is called, each element of the current gradient array OBJGRD or CJAC is initialised to a special value. On exit, any element that remains the value is estimated by finite differences.

3.4 MAIN FEATURES OF THE E04 UCF

At a solution of (3.1), some of the constraints will be active, i.e, satisfied exactly. An active simple bound constraint implies that the corresponding variable is fixed at its bound, and hence the variables are partitioned into fixed and free variables. Let C denote the mXn matrix of gradients of the active general linear and non-linear constraints. The number of fixed variables is denoted by n_{FX}, with n_{FR} ($N_{FR} = n - n_{FX}$) the number of free variables. The subscript 'FX'and 'FR'on a vector or matrix denote the vector or matrix composed of the components corresponding to fixed or free variables.

A point X is a first order Kuhn-Tucker point for (3.1) [28] if the following conditions hold:

(i) X is feasible;

(ii) There exists vector ξ and λ (the Lagrange multiplier vectors for the bound and general constraints) such that

$$g = C^T \lambda + \xi \tag{3.2}$$

Where g is the gradient of F(X) and $\xi_{\dot{J}} = 0$ if the $\dot{J}$ (th) variable is free.

(iii) The Lagrange multiplier corresponding to an inequality constraint active at its lower bound must be non-negative, and non-positive for an inequality constraint active at its upper bound.

Let Z denote a matrix whose columns form a basis for the set of vectors orthogonal to the rows of C_{FR} ; i.e. $C_{FR}\ Z = 0$. An equivalent statement of the condition (3.2) in terms of Z is:

$$Z^T g_{FR} = 0$$

The vector $Z^T g_{FR}$ is termed the projected gradient of F at X. Certai additional conditions must be satisfied in order for a first-order Kuhn-Tucker point to be a solution of (3.1) [28].

The basic structure of EO4UCF involves major and minor iterations. The major iterations generate a sequence of iterates $\{X_K\}$ that converge to X*, a first-order Kuhn-Tucker point of (3.1). At a typical major iteration, the new iterate $\bar{X}$ is defined by

$$\bar{X} \quad = \quad X + \alpha P \tag{3.3}$$

Where X is the current iterate, the non-negative scalar α is the step length, and P is the search direction. Also associated with each major iteration are estimates of the Lagrange multipliers and a prediction of the active set. The search direction P in (3.3) is the solution of a quadratic-programming subproblem of the form

$$\text{minimize} \quad g^T P + \frac{1}{2} P^T HP$$

$$\text{Subject to} \quad \bar{l} \leqslant \begin{bmatrix} P \\ A_L P \\ A_N P \end{bmatrix} \leqslant \bar{u} \tag{3.4}$$

Where g is the gradient of F at X, the matrix H is a positive definite quasi-Newton approximation to the Hessian of the Lagrangian function, and A_N is the Jacobian matrix of C evaluted at X. Let l in (3.1) be partioned into three sections: l_b, l_L and l_N, correspon-ding to the bounds, linear and non-linear constraints. The vector $\bar{l}$ in (3.4) is similarly partioned, and is defined as:

$$\bar{l}_B = l_B - X, \ \bar{l}_L = l_L - A_L X, \text{ and } \bar{l}_N = l_N - C,$$

Where C is the vector of non-linear constraints evaluated at X. The vector $\bar{u}$ is defined in an analogous fashion.

The estimated Lagrange multipliers at each major iteration are the Lagrange multipliers from the subproblem (3.4) (and similarl for the predicted active set). In EO4UCF (3.4) is solved using routine EO4NCF. Since solving a quadrative program as an iterative procedure, the minor iterations of EO4UCF are the iterations of routine EO4NCF.

Certain matrices associated with the QP subproblem are relevant in the major iterations. Let the subscripts 'FX' and 'FR' refers to the predicted fixed and free variables, and let C denote the mxn matrix of gradients of the general linear and non-linear constraints in the predicted active set. First, we have available the TQ factorization of C_{FR}:

$$C_{FR} \, Q_{FR} = (0 \ T) \qquad (3.5)$$

Where T is a non-singular mxm reverse-triangular matrix (i.e, $t_{ij} = 0$ if $i + J < n$), and the non singular $n_{FR} \times n_{FR}$ matrix Q_{FR} is the product of orthogonal transformation. Second, we have the upper triangular cholesky factor R of the transformed and re-ordered Hessian matrix

$$R^T R = H_Q \equiv Q^T \tilde{H} Q \tag{3.6}$$

Where $\tilde{H}$ is the Hessian H with rows and columns permuted so that the free variables are first, and Q is the nxn matrix

$$Q = \begin{bmatrix} Q_{FR} & \\ & I_{FR} \end{bmatrix} \tag{3.7}$$

with I_{FX} the identity matrix of order n_{FX}. If the columns of Q_{FR} are partioned so that the n_Z ($n_Z = n_{FR} - m$) columns of Z form

$$Q_{FR} = (Z\ Y),$$

a basis for the null space of C_{FR}. The matrix Z is used to compute the projected gradient $Z^T g_{FR}$ at the current iterate.

A theoretical characteristic of SQP methods is that the predicted active set from the QP subproblem (3.4) is identical to the correct active set in the neighbourhood of X^*. In E04UCF, this feature is exploited by using the QP active set from the previous iteration as a prediction of the active set for the next QP subproblem, which leads in practice to optimality of the subproblems in only one iteration as the solution is approached. Separate treatment of bound and linear constraints in E04UCF also saves computation in factorizing C_{FR} and H_Q.

Once P has been computed, the major iteration proceeds by determining a steplength α that produces a sufficient decrease in an augmented Lagrangian merit function. Finally, the approximation to the transformed Hessian matrix H_Q is updated using a modified BFCS quasi-Newton update to incorporate new curvature information obtained in the move from X to $\bar{X}$.

On entry to EO4UCF, an iterative procedure from routine EO4NCF is executed, starting with the provided initial point, to find a point that is feasible with respect to the bounds and linear constraints. If no feasible point exists for the bound and linear constraints, (3.4) has no solution and EO4UCF terminates. Otherwis the problem functions will thereafter be evaluted only at points that are feasible with respect to the bounds and linear constraints

In summary, the method of EO4UCF first determines a point that satisfies the bound and linear constraints. Thereafter, each iteration includes:

(a) The solution of a quadratic programming subproblem

(b) A line search with an augmented Lagrangian merit function, and

(c) A quasi-Newton update of the approximate Hessian of the Lagrangian function.

CHAPTER-4

RESULTS AND DISCUSSIONS

4.1 INPUT DATA

The objective functions formulated in chapter 2 has been optimized for various ambient conditions, fluid temperatures and capacities of the systems with the following design parameters.

Some of the parameters are common to all the three systems. They are:

Insulation conductivity	(K_I)	=	0.16 kJ/hr.m.C
Wind velocity	(v_a)	=	0.0 m/s
Emissivity of the surface	(ε)	=	0.05
Interest rate	(r)	=	10%
Life of the system	(L)	=	15 Years
Cost of insulating material	(C_2)	=	4000 Rs/m^3
Percentage installation factor	(F_1)	=	0.0
Percentage friction loss	(F_3)	=	0.0

Rest of the parameters for different cases are:

i. For Pipe System:

Water conductivity	(K_f)	=	2.09 kJ/hr.m.C
Kinematic viscosity of water	(ν_f)	=	1.55×10^{-6} m^2/s
Prandtle number	(P_{rf})	=	11.35
Roughness of steel pipe	(e)	=	4.5×10^{-4} m
Density of water	(ρ_w)	=	1000.0 Kg/m^3
Operating factor	(OF)	=	0.7
Efficiency of pump	(η_p)	=	0.8
Efficiency of motor	(η_m)	=	0.8
Head of the pump	(H_1)	=	45.0 m
Factor of safety	$(1+F_2)$	=	1.1
Cost of Refrigeration plant	(C_3)	=	12000 Rs./Ton
Cost of Pump with motor	(C_4)	=	25000 $Rs.min/m^3$

ii. For Duct System:

Air conductivity	(K_f)	=	0.091 kJ/hr.m.C
Kinematic viscosity of air	(ν_f)	=	15.37×10^{-6} m^2/s
Prandtle number	(P_{rf})	=	0.71
Density of air	(ρ_a)	=	1.21 Kg/m^3
Operating factor	(OF)	=	0.4
Efficiency of fan	(η_f)	=	0.8
Head of the fan	(H_2)	=	0.065 m

Factor of safety	$(1+F_2)$	=	1.1
Cost of duct material	(C_1)	=	190.0 Rs./m^2
Cost of Refrigeration plant	(C_3)	=	12000 Rs/Ton
Cost of fan	(C_4)	=	64.52 Rs.min/m^3

iii. For Heat Loss from Insulated Pipe:

Steam conductivity	(K_f)	=	0.290 kJ/hr.m.C
Kinematic viscosity of steam	(ν_f)	=	126.9×10^{-6} m^2/s
Density of steam	(ρ_s)	=	0.256 Kg/m^3
Prandtl number	(P_{rf})	=	1.08
Roughness of steel pipe	(e)	=	4.5×10^{-4} m
Operating factor	(OF)	=	0.7
Boiler maintenance cost factor	(F_4)	=	0.1
Cost of coal	(C_5)	=	1.5 Rs/Kg
Carolific value of fuel	(ΔH_s)	=	22453.7 KJ/Kg

4.2 PRESENTATION OF RESULTS

4.2.1 RESULTS IN TABULAR FORM

Table 4.1 shows the variation of optimal dimensions of the pipe system and its total cost, heat gain and electricity consumed per unit length of the pipe for two cases with the ambient temperat for various flow rate of water. In case I the total cost of the system is found after considering the maintenance cost also while in case II it is not considered. Temperature of water is taken to be constant at 280 K.

Evidently, dimensions of the pipe for case I are found to b lower than case II for the same flow rate and ambient temperature However, total cost, heat gain and electricity consumed turns out to be much higher in case I. As the insulation plays a great role in the total cost of the system, its lower thickness values are always desirable from maintenance and economy view points. Also, the inclusion of maintenance cost in total cost is more realistic one, therefore pipe dimensions as obtained in case I should always be choosen while erecting the pipe system.

Further, it is observed that changes in insulating layer thickness is comparatively more pronounced with greater temperatu potentials across the pipe wall than volume flow rate. Total cost heat gain from surroundings and electricity consumed are found to increase with volume flow rate and temperature potential across th wall. Even for the same temperature potential heat gain increases with volume flow rate. This is because of the higher exposed area at higher volume flow rate. From table 4.1 it is possible to calculate the optimum velocity of water flowing inside the pipe an it is found that it increases with volume flow rate and temperatu potential both. Its value lies between 1.15 to 1.53 m/s for volu flow rate between 1 to 25 m^3/min and ambient temperature between 17 to 47 C. From table 4.1 it is also possible to read out or interpolate the optimal dimensions for any volume flow rate betwee 1 to 25 m^3/min and ambient temperature betwen 17 C to 47 C.

Table 4.2 shows the similar variation for duct system. Temperature of the air flowing inside the duct is taken to be 280 For the same flow rate and temperature potential pipe dimensions a much higher than the duct dimensions. This is because of the hig

velocity of air flowing in the duct. For the same flow rate high velocity means lower inner diameter and hence lower exposed area heat transfer, so lower insulation thickness is required in duct systems than pipe systems for same flow rate. Optimum values of air velocity lies between 8 to 10.9 m/s for volume flow rate betw 10 to 2000 m^3/min and ambient temperature between 32 C to 47 C.

Table 4.3 shows the variation of insulation thickness alon with the total cost, heat loss, and coal consumed per unit length of pipe for two cases with the fluid temperature and volume flow rate for insulated pipe. Temperature of ambient air is taken to be constant at 310 K. Thickness of the insulation for case I are found to be lower than that in case II. However total cost, heat loss and coal consumed turn out to be higher in case I. As temperature of the superheated steam increases all values of the table also increase. If we choose a pipe of higher diameter for same flow rate, all the values listed in table also increases.

Table 4.4 shows the variation of optimal dimensions along with total cost heat gain and, electricity consumed per unit leng of pipe for two cases, with the ambient temperature for various water temperatures. Volume flow rate of water is taken to be con at 6 m^3/min. It is observed that for the same temperature potent optimal dimensions, total cost, heat gain and electricity changes are almost the same and as the temperature potential increases in diameter of the pipe decreases but rest of the quantities increase It is the temperature potential which affects the various quantie listed in the table 4.4 not the absolute value of these temperatu Table 4.5 shows the similar variations for theduct system. Here volume flow rate is taken to be 500 m^3/min.

4.2.2 SENSITIVITY ANALYSIS

Figures 4.1 to 4.6 shows the effect of varying the design variables about the optimal point on the positive and negative sic by 50% in steps of 10%. It is observed that for the pipe and duct systems the total cost as well as electricity consumption is more sensitive to the inner diameter and less sinsitive to insulation thickness whereas refrigeration loss is equally sensitive to both Figure 4.7 shows the variation in the total cost, heat gain and cc consumption with the insulation thickness for the heat loss from the insulated pipe.

4.2.3 ENERGY AS THE CRITERIA FOR SELECTING THE SYSTEM COMPONENTS

Table 4.6 shows the influence of changing the insulation thickness (from the optimum point) on the total cost, electricity consumed and refrigeration loss for pipe system. While the inner diameter is fixed at optimum value. It is seen that if we increas the insulation thickness by 20%, 30% and 40%, the total cost increases only by 0.473%, 1.003% and 1.690% only, whereas it gives saving in precious electrical energy by 5.55%, 7.78% and 9.75%, respectively. Since the marginal energy saving at the higher insulation thickness is not increasing significantly, we can't go on adding insulation thickness indefinitely.30 to 40% more insulat looks to a reasonable choice to save 7.8 to 9.7% electricity.

Table 4.7 shows the influence of changing the inner diamete (from the optimum point) on the total cost and electricity consume for pipe system while the insulation thickness is fixed at optimur value. It is seen from table 4.7 that if we increase the inner

diameter by 3.25%, 6.50% and 9.75%, the total cost increases by 0.306%, 1.146% and 2.421% whereas it gives saving in electricity by 6.42%, 11.52% and 15.55%. Unlike the case of insulation thickn it reaches a minimum level of energy consumption (26.86% saving) around a point which is 40% higher than the optimum inner diameter But at this point total cost rises by 25% (too high value). A glan at the table 4.7 shows that we should increase the inner diameter by 10% more than optimum this would save 16% electricity at the co of megre 2.5% more investment in terms of money. Thus around 25% electricity could be saved at the cost of 3.5% additional money.

Similarly, table 4.8 and 4.9 shows the influence of the changing insulation thickness and inner diameter respectively for the duct system. If we increase the insulation thickness by 20%, 30%, 40% and 50%, the total cost increases only by 0.618%, 1.315%, 2.219% and 3.300% whereas it gives saving in electrical energy by 7.5%, 10.62, 13.42%, 15.94%, respectively. Here also we cannot go on adding insulation thickness indefinitely as increase in cost is more than saving in electricity at higher values of insulation thickness. Around 30% more insulation may be added to save 13.42% of electricity. If we increase the inner diameter of duct by 5%, 10% and 15%, the total cost increases by 0.472%, 1.701%, 3.479% wh as it gives saving in electricity by 3.87%, 6.19% and 7.40%. Here also unlike the case of insulation thickness it reaches a minimum level of energy consumption (7.78% saving) around 20% increase in inner diameter and it gives increase in the cost by 5.655% only. So by selecting the duct dimensions judicially on 12%, 15%, 18%,2

22% electrical energy could be saved at the cost of 1.1%, 1.8%, 2.7 3.8%, 5.0% respectively. From these data is quite clear that every unit saved is becoming costier and costier.

Similarly, table 4.10 shows the influence of the changing insulation thickness for the heat loss through insulated pipe. If we increase the insulation, thickness by 20%, 30% and 40%, the tota cost increases by 1.5%, 3.18% and 5.36% whereas it gives saving in electricity by 11.82%, 16.47% and 20.50%, respectively. So 15% electricity could be saved just by allowing an increase in the toal cost by 2.5%.

In view of the rising cost and greater demand of electricity it is desirable to select the dimensions higher than the optimum value in order to save more and more energy within the tolerable limit of additional cost incurred.

4.3 EFFECTS OF VARIOUS PARAMETERS ON OPTIMAL DIMENSIONS

4.3.1 EFFECTS OF WIND VELOCITY SURFACE EMISSIVITY AND INSULATING MATERIAL (THERMAL PARAMETERS)

Effect of wind velocity on optimum dimensions, total cost, refrigeration loss and surface temperature are shown in table 4.11 for the pipe system for a typical situation. It is seen that insulation thickness, surface temperature, total cost and refrigeration loss increases with wind velocity but variation in inner diameter is neglible. Refrigeration loss due to radiation decrease as wind velocity increases because of higher surface temperatures but due to higher value of outer convection heat transfer coefficie

at higher wind velocity, refrigeration loss due to convection increases. The net effect of these two losses is to give the higher refrigeration losses at the higher wind velocities. Table 4.12 shows similar results for duct system as well.

Table 4.13 shows the effect of wind velocity for heat loss through insulated pipe. It is seen that insulation thickness, total cost, heat loss and heat loss due to radiation increase as wind velocity increases but surface temperature decreases.

Effect of surface emissivity on optimum dimensions, total cost, refrigeration loss and surface temperature is shown in table 4.14 for the pipe system. It is is seen that insulation coverings with low emissivity such as Aluminium produces lower values of insulation thickness, total cost, refrigeration loss and surface temperatures than coverings with high emissivities. Refrigeration loss due to variation in inner diameter with emissivity is neglible. Radiation heat from surface increases as the surface emissivity is increased but this increase is not in the same proportion as surface emissivity because surface temperature also increases. The outer heat transter coefficient decreases as emissivity is increased, hence convection losses also decreases. The net effect of these two losses is to give almost same total refrigeration loss. Therefore, use of higher emissivity surfaces do not alter the inner diameter and refrigeration loss much but total cost increases mainly because of the higher value of the insulation thickness.

Table 4.15 shows similar results for duct system. Table 4.16 shows the effect of surface emissivity for heat loss through insulated pipes. It is seen here that insulation thickness, total cost and

heat loss increases slightly with emissivity but surface temperature decreases. Heat loss due to radiation also increases with emissivit

Table 4.17 shows the effect of insulating materials choosen on the optimal dimensions, total cost, electricity consumed and refrigeration loss for the pipe system. It is seen that insulating materials with lower thermal conductivity, such as thermocole render lower insulation thicknesses and refrigeration losses than materials with higher conductivities. However total cost and electricity consumed need not necessarily be lower for lower conductivity materials. This is because, cost of materials also changes with conductivities. If there is significant difference in total cost and energy consumed for different choices of insulating materials, then depending on the criteria we choose one having either lowest total cost or lowest electricity consumed or one having both lowest Variation in inner diameter is insignificant. Table 4.18 and 4.19 shows the similar variations for the duct system and heat loss through insulated pipe respectively.

4.3.2 EFFECTS OF OPERATING FACTOR

Table 4.20 shows the effect of operating factor on optimal dimensions and its total cost and electricity consumed for pipe system for a typical situation. It is seen that optimal dimensions total cost and electricity vary significantly with operating factor and their values are lower for lower operating factor. This sugges that optimum velocity is dependent on the number of hours per day o year for which the system is in operation. Tables 4.21 and 4.22 shows the similar results for the duct system and heat loss through insulated pipe, respectively.

4.3.3 EFFECTS OF COST PARAMETERS

Tables 4.23 to 4.30 show the effects of pump/fan, cost of refrigeration plant, and cost of pipe/duct installation on optimal dimensions, total cost and electricity for pipe and duct systems. It is seen that when we also consider the pipe or duct installation cost in total cost function, optimal dimensions decreases significantly. This effect is more pronounced in case of insulation thickness. Total cost and electricity consumed increases when we also consider this cost. Total cost increases mainly because of this additional cost whereas electricity consumption increases because of lower dimensions.

If we consider the cost of refrigeration plant, optimal dimensions are found to be higher than when we do not consider this cost for the pipe system. For the duct system insulation thickness decreases and inner diameter increases when refrigeration plant cost is not considered. In both the systems total cost decreases and electricity consumption increases significantly when this cost is not considered. Insulation thickness is more pronounced to this effect. Hence ignoring this cost will produce wrong optimal dimensions particularly insulation thickness. Decrease in total cost is deceptive because in actual practice we have to pay for the refrigeration plant. Actual decrease in total cost is only because of the lower dimensions used now.

Inner diameter decreases slightly and insulation thickness remain practically unaffected when pump/fan cost is not considered than when it is considered. Total cost decreases but electricity consumption increases when this cost is not considered.

If we do not consider both these costs, then optimum dimensions and total cost decreases but electricity consumption increases. As mentioned earlier decrease in total cost in actual practice will not be as much as shown in the tables 4.29 & 4.30 . So by ignoring any one or both of these costs would mean wastage of more electricity.

Table 4.31 shows the effect of duct thickness on optimal dimensions. The inner diameter is found to increase slightly as the duct thickness decreases. Total cost also decreases because of the lower cost for thin sheets. If a particular application does not pose any restriction on the thickness of sheet than we should choose the one having minimum thickness.

Table 4.32 shows the effect of boiler maintenance cost on the insulation thickness, total cost and coal consumption. Optimum insulation thickness is found to be the largest and coal consumption and total cost is the least when this cost is not included. Thus the exclusion of this cost will give the higher value of the insulat thickness and conservative value of coal consumption.

The units of the various parameters in the following tables are as follows:

d_1	(mm)	V	(m^3/min)	K_I	(w/mK)
$\frac{d_2-d_1}{2}$	(mm)	T_i	(K)	C_1	(Rs/m^2)
C_T	(Rs./m)	T_a	(K)	C_2	(Rs./m^3)
$\dot{Q}$	(w)	T_s	(K)	C_3	(Rs/Ton)
R_C	(Kg/m)	V_a	(m/s)	C_4	(Rs/m^3/min)
R_E	(Kwh)	h_o	(w/m^2K)		

Table 4.1: Comparison of optimal dimensions of a pipe system having glass wool insulation layer.

V	T_a	Case 1 with maintenance cost					Case 2 without maintenance cost				
		d_1	$\frac{d_2-d_1}{2}$	C_T	$\dot{Q}$	R_E	d_1	$\frac{d_2-d_1}{2}$	C_T	$\dot{Q}$	R_E
1	290	140	20	790	7	450	140	27	721	6	419
	300	137	32	943	11	598	138	41	834	9	546
	310	135	40	1069	14	722	136	51	929	12	655
	320	133	47	1181	17	834	135	59	1014	15	753
6	290	315	20	2285	13	1141	314	28	2133	11	1083
	300	310	34	2574	20	1406	311	45	2340	17	1303
	310	307	44	2804	25	1622	308	57	2508	21	1485
	320	304	53	3005	30	1811	306	67	2655	26	1646
10	290	397	20	3127	16	1514	396	28	2935	13	1445
	300	391	35	3478	24	1832	392	46	3185	20	1707
	310	388	45	3756	30	2089	389	58	3385	25	1921
	320	385	54	3997	36	2313	387	69	3561	30	2111
25	290	600	20	5535	23	2553	598	28	5242	19	2464
	300	590	35	6042	34	3003	593	47	5598	28	2829
	310	589	46	6439	43	3360	583	60	5881	35	3122
	320	585	55	8780	51	3668	586	71	6125	42	3378

Table 4.2: Comparison of optimal dimensions of a duct system having glass wool insulation layer.

V	T_a	Case 1 with maintenance cost					Case 2 without maintenance cost				
		d_1	$\frac{d_2-d_1}{2}$	C_T	$\dot{Q}$	R_E	d_1	$\frac{d_2-d_1}{2}$	C_T	$\dot{Q}$	R_E
	305	162	20	430	11	270	164	26	344	10	242
10	310	158	25	492	13	317	161	32	388	12	283
	315	155	29	548	15	360	158	36	428	13	321
	320	152	33	600	17	400	156	41	465	15	357
	305	493	20	1191	28	719	499	27	956	24	635
	310	481	26	1349	33	829	489	34	1065	28	729
100	315	471	31	1488	37	925	480	40	1162	32	813
	320	464	35	1614	40	1013	474	45	1249	35	889
	305	1077	18	2475	58	1489	1087	26	1999	49	1309
	310	1048	25	2802	66	1705	1063	33	2220	56	1491
500	315	1027	30	3085	74	1893	1045	40	2414	63	1651
	320	1010	35	3338	81	2063	1031	46	2587	69	1795
	305	2112	16	4663	111	2828	2126	24	3793	93	2482
	310	2053	23	5291	126	3231	2079	32	4214	106	2818
2000	315	2009	29	5831	139	3580	2042	39	4579	118	3113
	320	1975	34	6311	151	3892	2013	45	4905	128	3375

Table 4.3: Comparison of optimal insulation thickness for heat loss from insulated pipe having glass wool insulation layer.

V	d_1	T_i	Case 1 with maintenance cost				Case 2 without maintenance cost			
			$\frac{d_2-d_1}{2}$	C_T	$\dot{Q}$	R_C	$\frac{d_2-d_1}{2}$	C_T	$\dot{Q}$	R_C
		573	106	3993	124	1141	142	3397	102	915
		673	123	4713	155	1477	164	3993	129	1203
100	300	773	138	5370	184	1786	182	4540	154	1469
		873	150	5984	212	2078	198	5053	178	1721
		973	161	6566	238	2357	212	5541	200	2963
		573	110	5525	150	1507	148	4749	131	1217
		673	129	6372	186	1885	172	5437	152	1535
200	400	773	144	7144	220	2235	191	6069	181	1832
		873	157	7864	252	2566	208	6661	208	2113
		973	169	8590	281	2884	223	7224	234	2383
		573	117	8266	201	1831	159	7115	159	1408
		673	137	9425	247	2331	184	8044	198	1826
400	600	773	153	10470	290	2787	206	8884	233	2208
		873	168	11430	329	3212	224	9862	267	2565
		973	181	12330	368	3615	241	10400	298	2905

Table 4.4: Comparison of optimal dimensions for various combinations of fluid and ambient temperature for $V = 6\ m^3/min$.

T_i	T_a	Case 1 with maintenance cost					Case 2 without maintenance cost				
		d_1	$\frac{d_2-d_1}{2}$	C_T	$\dot{Q}$	R_E	d_1	$\frac{d_2-d_1}{2}$	C_T	$\dot{Q}$	R_E
	290	312	28	2439	16	1282	312	37	2243	14	1200
	300	308	40	2693	23	1518	309	51	2427	19	1397
275	310	306	49	2907	28	1718	307	62	2583	24	1567
	320	303	56	3097	32	1899	305	71	2723	28	1721
	290	315	20	2285	13	1141	314	28	2133	11	1083
	300	310	34	2574	20	1406	311	45	2340	17	1303
280	310	307	44	2804	25	1622	308	57	2508	21	1485
	320	304	53	3005	30	1811	306	67	2655	26	1646
	290	319	9	2090	8	966	317	15	1995	7	939
	300	312	28	2440	16	1283	314	37	2243	14	1200
285	310	308	40	2694	23	1518	309	52	2428	19	1397
	320	305	49	2908	28	1719	307	62	2584	24	1568

Table 4.5: Comparison of optimal dimensions for various combinations of fluid and ambient temperature. For $\dot{V} = 500\ m^3/min$.

T_i	T_a	Case 1 with maintenance cost					Case 2 without maintenance cost				
		d_1	$\frac{d_2-d_1}{2}$	C_T	$\dot{Q}$	R_E	d_1	$\frac{d_2-d_1}{2}$	C_T	$\dot{Q}$	R_E
	305	1048	25	2799	66	1704	1063	33	2219	56	1490
	310	1027	30	3083	74	1892	1046	40	2403	63	1646
285	315	1010	35	3336	81	2062	1031	46	2586	69	1794
	320	996	39	2567	87	2217	1019	51	2745	74	1927
	305	1077	18	2475	58	1489	1087	26	1999	49	1309
	310	1048	25	2802	66	1705	1063	33	2220	56	1491
290	315	1027	30	3085	74	1893	1045	40	2414	63	1651
	320	1010	35	3338	81	2063	1031	46	2587	69	1795
	305	1120	11	2079	47	1231	1120	17	1733	40	1091
	310	1076	19	2478	58	1490	1087	26	2001	49	1309
295	315	1048	25	2804	68	1706	1063	34	2222	56	1492
	320	1027	30	3088	74	1894	1045	40	2415	63	1651

Table 4.6: Sensitivity analysis of pipe system with respect to insulation thickness.

d_1	$\frac{d_2-d_1}{2}$	C_T	R_E	$\dot{Q}$	% age change in $\frac{d_2-d_1}{2}$	% age change in C_T	% age change in R_E	% age change in $\dot{Q}$	T_s
307	36	2822	1746	29.3	-20	0.631	7.67	16.10	302.0
307	40	2808	1678	27.1	-10	0.145	3.50	7.34	302.7
307	44	2804	1621	25.2	0.0	0.0	0.0	0.0	303.2
307	48	2807	1573	23.6	10	0.126	-2.98	- 6.25	303.6
307	53	2817	1532	22.3	20	0.473	-5.55	-11.64	304.0
307	57	2832	1495	21.1	30	1.003	-7.78	-16.34	304.3
307	62	2852	1463	20.0	40	1.690	-9.75	-20.47	304.6
307	66	2875	1435	19.1	50	2.512	-11.50	-24.13	304.9

Table 4.7: Sensitivity analysis for pipe system with respect to inner diameter.

d_1	$\frac{d_2-d_1}{2}$	C_T	R_E	$\dot{Q}$	% age change in $\frac{d_2-d_1}{2}$	% age change in C_T	% age change in R_E	% age change in $\dot{Q}$	T_s
287	44	2847	1918	23.8	- 6.50	1.534	18.28	- 5.39	303.2
297	44	2814	1753	24.5	- 3.25	0.354	8.10	- 2.69	303.2
307	44	2804	1622	25.2	0.0	0.0	0.00	00.00	000.0
317	44	2812	1517	25.9	3.25	0.306	- 6.42	2.69	303.1
327	44	2836	1435	26.6	6.50	1.146	-11.52	5.37	303.1
337	44	2872	1369	27.2	9.75	2.421	-15.55	8.05	303.0
347	44	2918	1318	27.9	13.0	4.053	-18.71	10.72	303.0
357	44	2972	1278	28.6	16.25	5.983	-21.18	13.39	303.0

Table 4.8: Sensitivity analysis for duct system with respect to insulation thickness.

d_1	$\frac{d_2-d_1}{2}$	C_T	R_E	$\dot{Q}$	% age change in $\frac{d_2-d_1}{2}$	% age change in C_T	% age change in R_E	% age change in $\dot{Q}$	T_s
1027	24	3110	2079	84	-20	0.803	9.82	14.36	305.3
1027	27	3091	1980	79	-10	0.187	4.56	6.66	305.9
1027	30	3085	1893	74	0	0.000	0.00	0.00	306.4
1027	33	3090	1818	70	10	0.164	-3.98	-5.83	306.8
1027	37	3104	1751	66	20	0.618	-7.50	-10.97	307.2
1027	40	3126	1692	62	30	1.315	-10.62	-15.54	307.6
1027	43	3154	1639	59	40	1.219	-13.42	-19.64	307.9
1027	46	3187	1591	57	50	3.300	-15.94	-23.32	308.2

Table 4.9: Sensitivity analysis for duct system with respect to inner diameter.

d_1	$\frac{d_2-d_1}{2}$	C_T	R_E	$\dot{Q}$	% age change in $\frac{d_2 - d_1}{2}$	% age change in C_T	% age change in R_E	% age change in $\dot{Q}$	T_s
927	30	3168	2180	68	9.74	2.700	15.16	- 8.15	306.4
977	30	3103	2008	71	-4.87	0.593	6.04	- 4.04	306.4
1027	30	3085	1893	74	0.00	0.000	0.00	0.00	306.4
1077	30	3100	1820	77	4.87	0.472	-3.87	3.99	306.3
1127	30	3138	1776	80	9.74	1.701	-6.19	7.92	306.3
1177	30	3192	1753	83	14.61	3.479	-7.40	11.79	306.3
1227	30	3260	1746	85	19.48	5.655	-7.78	15.69	306.3
1277	30	3336	1750	88	24.35	8.125	-7.56	19.37	306.3
1327	30	3419	1763	91	29.22	10.812	-6.90	23.07	306.2

Table 4.10: Sensitivity analysis for heat loss through insulated pipe with respect to insulation thickness.

d_1	$\frac{d_2-d_1}{2}$	C_T	R_C	$\dot{Q}$	% age change in $\frac{d_2-d_1}{2}$	% age change in C_T	% age change in R_C	% age change in $\dot{Q}$	T_s
300	120	6105	2435	247	-20	2.024	17.18	16.35	345.7
300	135	6012	2238	228	-10	0.469	7.72	7.35	342.3
300	150	5984	2078	212	0	0.000	0.00	0.00	339.6
300	165	6008	1945	199	10	0.398	- 6.40	- 6.09	337.2
300	180	6074	1832	188	20	1.493	-11.82	-11.25	335.3
300	195	6174	1736	179	30	3.173	-16.47	-15.68	333.6
300	210	6305	1652	171	40	5.356	-20.50	-19.52	332.1
300	225	6462	1578	164	50	7.979	-24.04	-22.89	330.8

Table 4.11: Effect of wind velocity on optimal dimensions for the pipe system.

V_a	d_1	$\frac{d_2-d_1}{2}$	C_T	$\dot{Q}$	T_s	h_o	% age radiation heat gain
0.0	307	44	2804	25	303.16	2.6	11.1
0.5	306	48	2845	26	305.07	3.7	8.1
2.0	306	51	2878	26	306.64	3.7	5.6
5.0	305	52	2898	26	307.55	7.9	4.1

Table 4.12: Effect of wind velocity on optimal dimensions for the duct system.

V_a	d_1	$\frac{d_2 - d_1}{2}$	C_T	$\dot{Q}$	T_s	h_o	% age radiation heat gain
0.0	1027	30	3085	74	306.37	2.2	13.6
0.5	1017	35	3210	74	308.73	3.1	10.0
2.0	1008	38	3313	74	310.69	4.7	6.9
5.0	1005	40	3373	75	311.85	6.6	5.1

Table 4.13: Effect of wind velocity on insulated thickness for the heat loss through insulated pipe.

V_a	d_1	$\frac{d_2 - d_1}{2}$	C_T	$\dot{Q}$	T_C	h_o	% age radiation heat loss
0.0	300	150	5984	212	339.55	3.4	10.3
0.5	300	152	6033	213	331.30	4.9	7.1
2.0	300	154	6073	214	324.47	7.4	4.7
5.0	300	155	6096	215	320.53	10.2	3.7

Table 4.14: Effect of surface emissivity on optimal dimensions for the pipe system.

ε	d_1	$\frac{d_2 - d_1}{2}$	C_T	$\dot{Q}$	T_s	h_o	% age radiation heat gain
0.0	307	44	2790	25	302.52	2.7	0.0
0.05	307	44	2804	25	303.16	2.6	11.1
0.10	307	46	2819	25	303.84	2.5	20.6
0.2	306	47	2836	26	304.64	2.5	34.9
0.5	306	50	2873	26	306.35	2.2	59.9

Table 4.15: Effect of surface emissivity on optimal dimensions for the duct system.

ε	d_1	$\frac{d_2 - d_1}{2}$	C_T	$\dot{Q}$	T_s	h_o	% age radiation heat gain
0.0	1032	29	3028	73	305.36	2.2	0.0
0.05	1027	30	3085	74	306.37	2.2	13.6
0.10	1021	32	3190	74	307.49	2.1	25.2
0.20	1016	34	3203	74	308.55	2.0	40.71
0.50	1007	38	3323	75	310.83	1.8	65.93

Table 4.16: Effect of surface emissivity on optimal dimensions for the heat loss through insulated pipe.

ϵ	d_1	$\frac{d_2 - d_1}{2}$	C_T	$\dot{Q}$	T_C	h_o	% age radiation heat loss
0.0	300	149	5968	212	342.21	3.5	0.0
0.05	300	150	5984	212	339.55	3.4	10.0
0.10	300	150	5997	213	337.30	3.3	18.8
0.20	300	151	6019	213	333.68	3.2	32.1
0.50	300	153	6059	214	326.91	3.0	55.4

Table 4.17: Effect of insulating materials on optimal dimensions for the pipe system.

Insulating material K_I	C_2	d_1	$\frac{d_2 - d_1}{2}$	C_T	R_E	$\dot{Q}$
Thermocle (0.033)	6000	306	32	2812	1617	25
	4500	308	38	2708	1515	22
Glasswool (0.044)	4000	307	44	2804	1622	25
Woolen cloth (0.189)	1000	306	14	3175	2111	41

Table 4.18 : Effect of insulating materials on optimal dimensions for the duct system.

Insulating material K_I	C_2	d_1	$\frac{d_2 - d_1}{2}$	C_T	R_E	$\dot{Q}$
Thermocole (0.033)	6000	1022	21	3153	1946	76
	4500	1035	26	2938	1759	68
glasswool (0.044)	4000	1027	30	3085	1893	74
woolen cloth (0.189)	1000	1027	108	3318	2174	90

Table 4.19: Effect of insulating materials on optimal dimensions for the heat loss through insulated pipe.

Insulating material K_I	C_2	d_1	$\frac{d_2 - d_1}{2}$	C_T	R_C	$\dot{Q}$
Thermocole (0.033)	6000	300	112	5776	1907	196
	4500	300	127	5322	1737	179
glasswool (0.044)	4000	300	150	5984	2078	212
woolen cloth (0.189)	1000	300	449	9971	4472	445

Table 4.20: Effect of operating factor on optimal dimensions for the pipe system.

OF	d_1	$\frac{d_2 - d_1}{2}$	C_T	R_E
0.4	288	36	2809	1519
0.5	295	39	2526	1319
0.6	301	42	2670	1475
0.7	307	44	2804	1622

Table 4.21: Effect of operating factor on optimal dimensions for the duct system.

OF	d_1	$\frac{d_2 - d_1}{2}$	C_T	R_E
0.3	1006	27	2809	1519
0.4	1027	30	3085	1893
0.5	1044	33	3344	2236

Table 4.22: Effect of operating factor on optimal dimensions for the heat loss through insulated pipe.

OF	d_1	$\frac{d_2 - d_1}{2}$	C_T	R_C
0.5	300	130	5049	1645
0.6	300	140	5529	1867
0.7	300	150	5984	2078

Table 4.23: Effect of percentage pipe installation cost factor on optimal dimensions for the pipe system.

F_1	d_1	$\frac{d_2 - d_1}{2}$	C_T	R_E
0.0	307	44	2804	1622
0.3	296	38	3275	1837
0.6	288	34	3716	2033

Table 4.24: Effect of percentage pipe installation factor on optimal dimensions for the duct system.

f_1	d_1	$\frac{d_2 - d_1}{2}$	C_T	R_E
0.0	1027	30	3085	1893
0.3	1003	25	3468	2097
0.6	984	21	3811	2274

Table 4.25: Effect of cost of refrigeration plant for the pipe system.

C_3	d_1	$\frac{d_2 - d_1}{2}$	C_T	R_E
0	306	37	2578	1737
12000	307	45	2804	1622

Table 4.26: Effect of cost of Refrigeration plant for the duct system.

C_3	d_1	$\frac{d_2 - d_1}{2}$	C_T	R_E
0	1049	20	2452	2216
12000	1027	30	3085	1893

Table 4.27: Effect of the cost of pump on optimal dimensions for the pipe system.

C_4	d_1	$\frac{d_2 - d_1}{2}$	C_T	R_E
0	304	44	2717	1657
25000	307	44	2804	1622

Table 4.28: Effect of the cost of fan on optimal dimensions for the duct system.

C_4	d_1	$\frac{d_2 - d_1}{2}$	C_T	R_E
0	993	30	2994	1961
64.52	1027	30	3085	1893

Table 4.29: Combined effect of the cost of Refrigeration plant and pump on optimal dimensions for the pipe system.

C_3	C_4	d_1	$\frac{d_2 - d_1}{2}$	C_T	R_E
0	0	303	37	2545	1775
12000	25000	307	49	2804	1622

Table 4.30: Combined effect of the cost of Refrigeration plant and fan on optimal dimensions for the duct system.

C_3	C_4	d_1	$\frac{d_2 - d_1}{2}$	C_T	R_E
0	0	1009	20	2369	2269
12000	64.52	1027	30	3085	1893

Table 4.31: Comparison of optimal dimensions for ducts of different thickness

Thickness of duct material (gauge)	d_1	$\frac{d_2 - d_1}{2}$	C_T	R_E
24	1038	30	2939	1875
22	1033	30	3004	1883
20	1029	30	3053	1889
18	1027	30	3085	1893

Table 4.32: Cost of boiler maintenance included.

F_4	d_1	$\frac{d_2 - d_1}{2}$	C_T	R_C
0.0	300	153	5912	2051
0.1	300	150	5984	2078
0.3	300	145	6123	2131

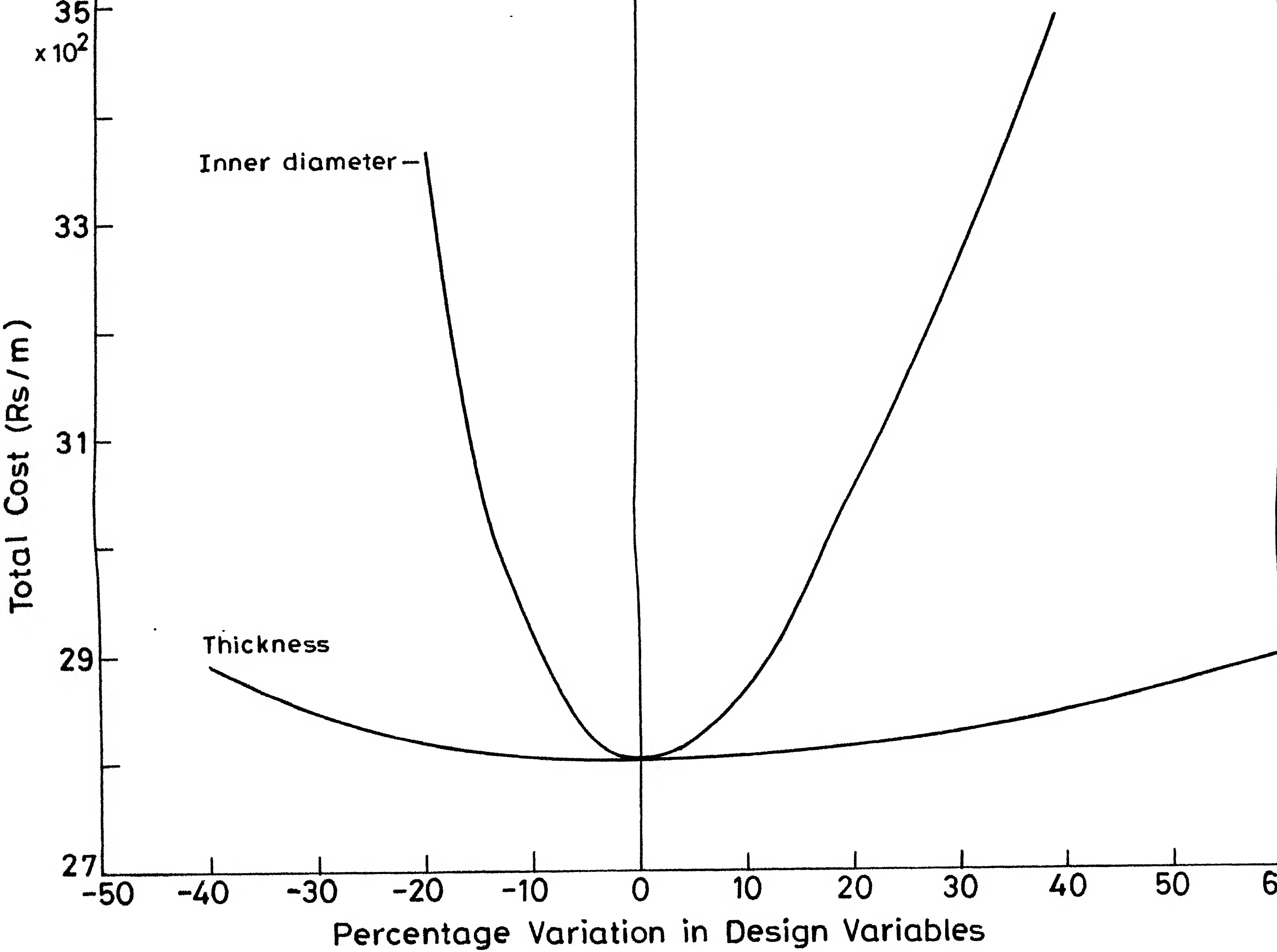

Fig. 4.1 Sensitivity analysis of total cost.

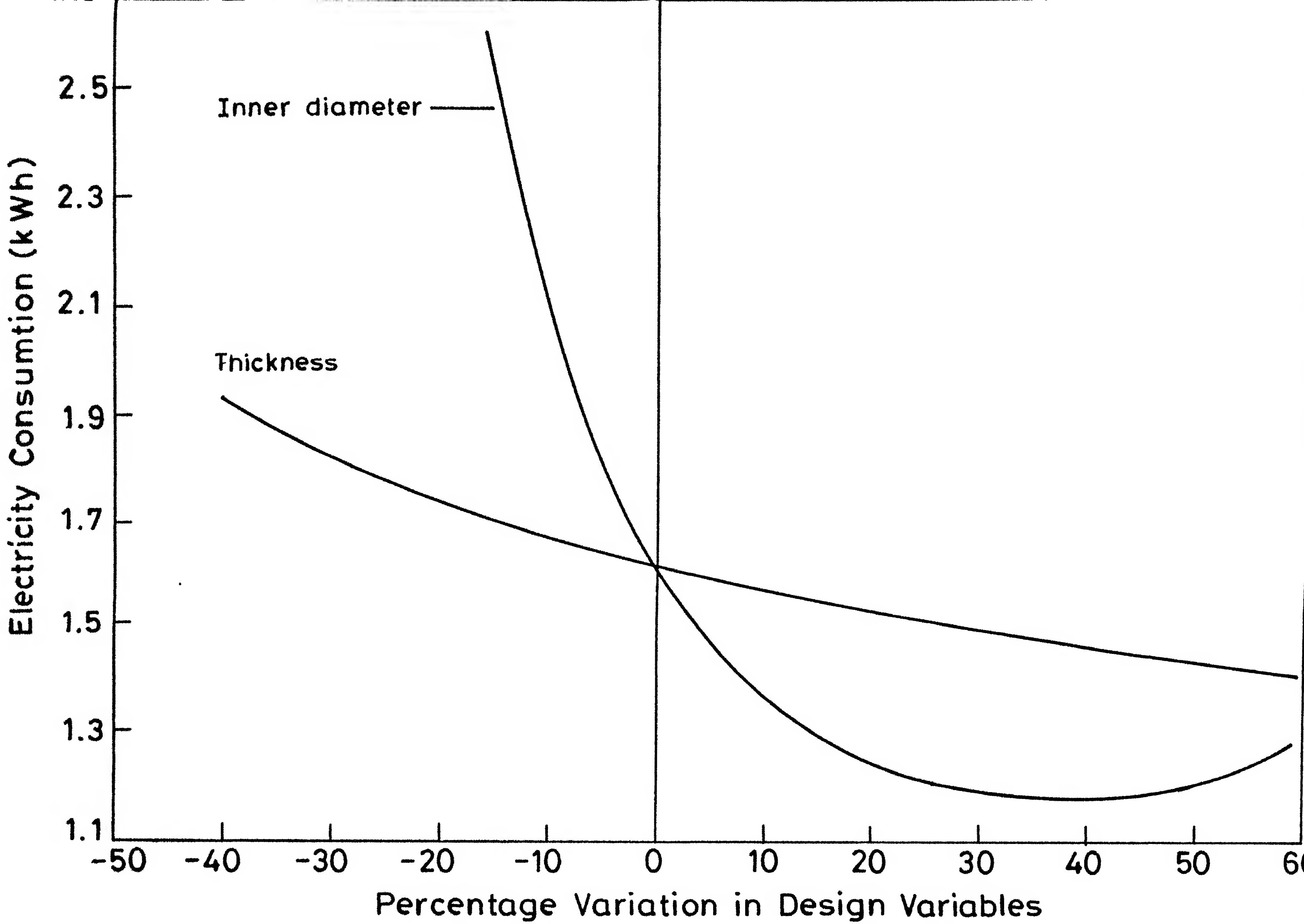

Fig. 4.2 Sensitivity analysis of electricity consumption.

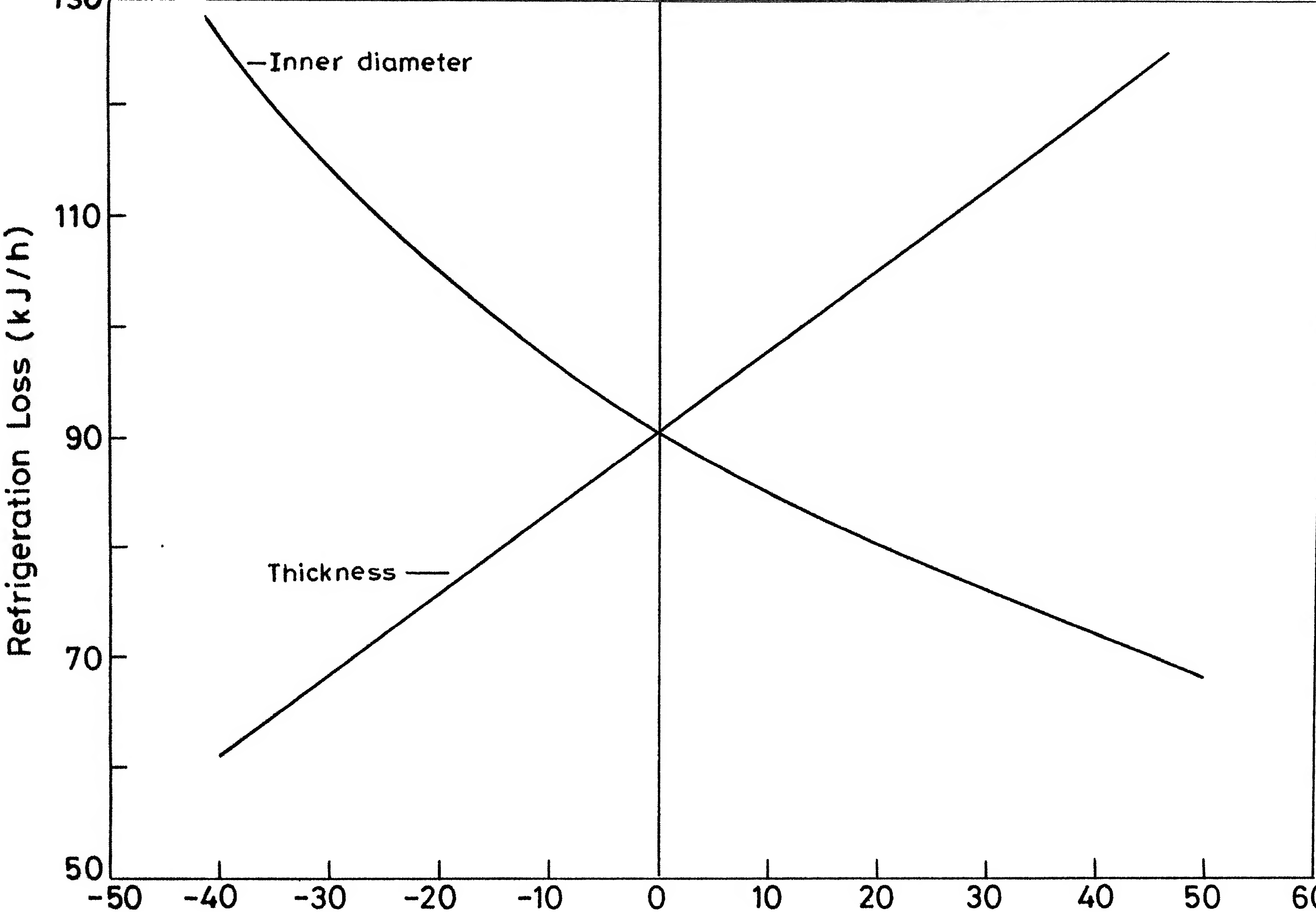

Fig. 4.3 Sensitivity analysis of refrigeration loss.

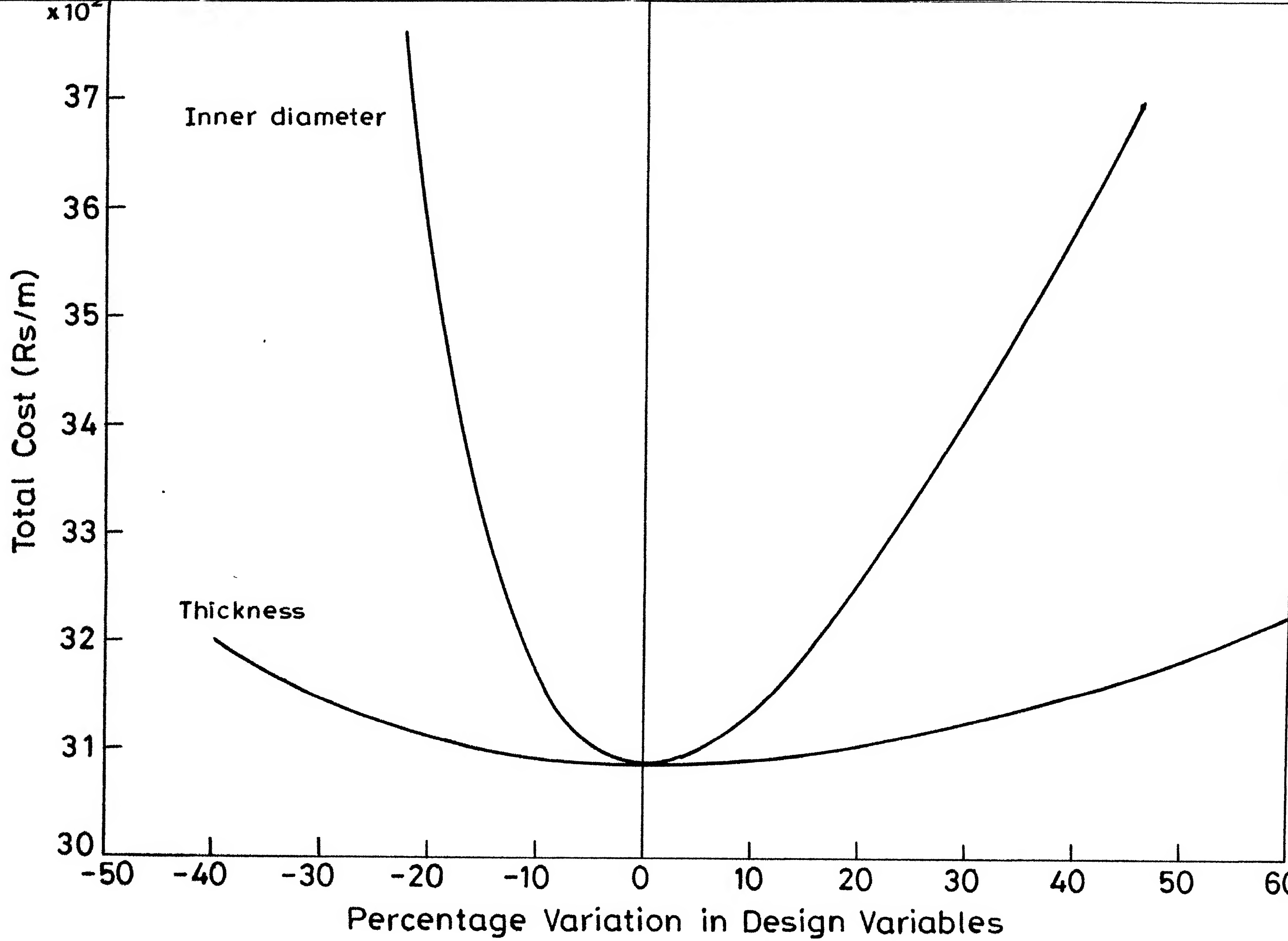

Fig. 4.4 Sensitivity analysis of total cost.

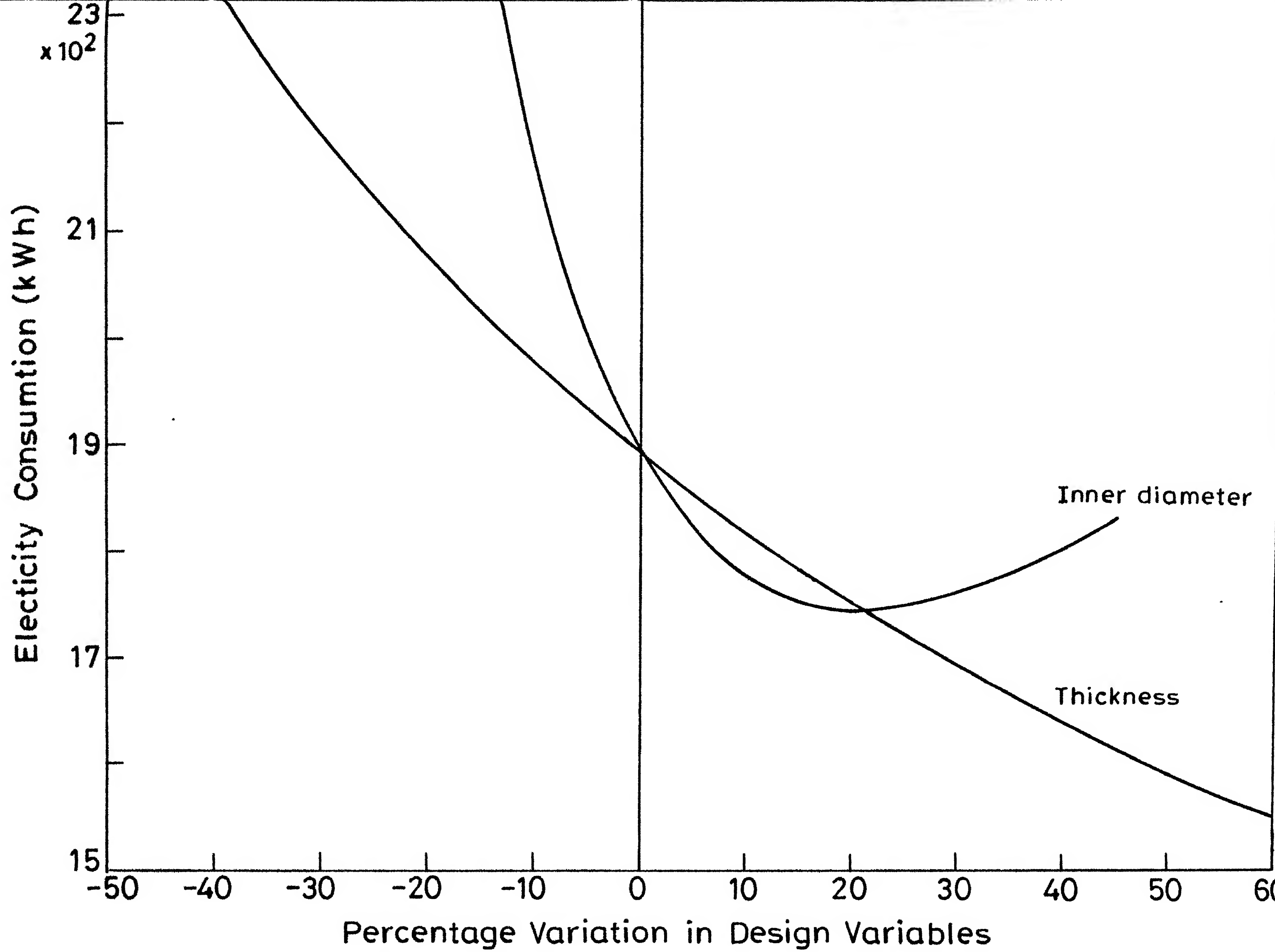

Fig. 4.5 Sensitivity analysis of electricity consumption.

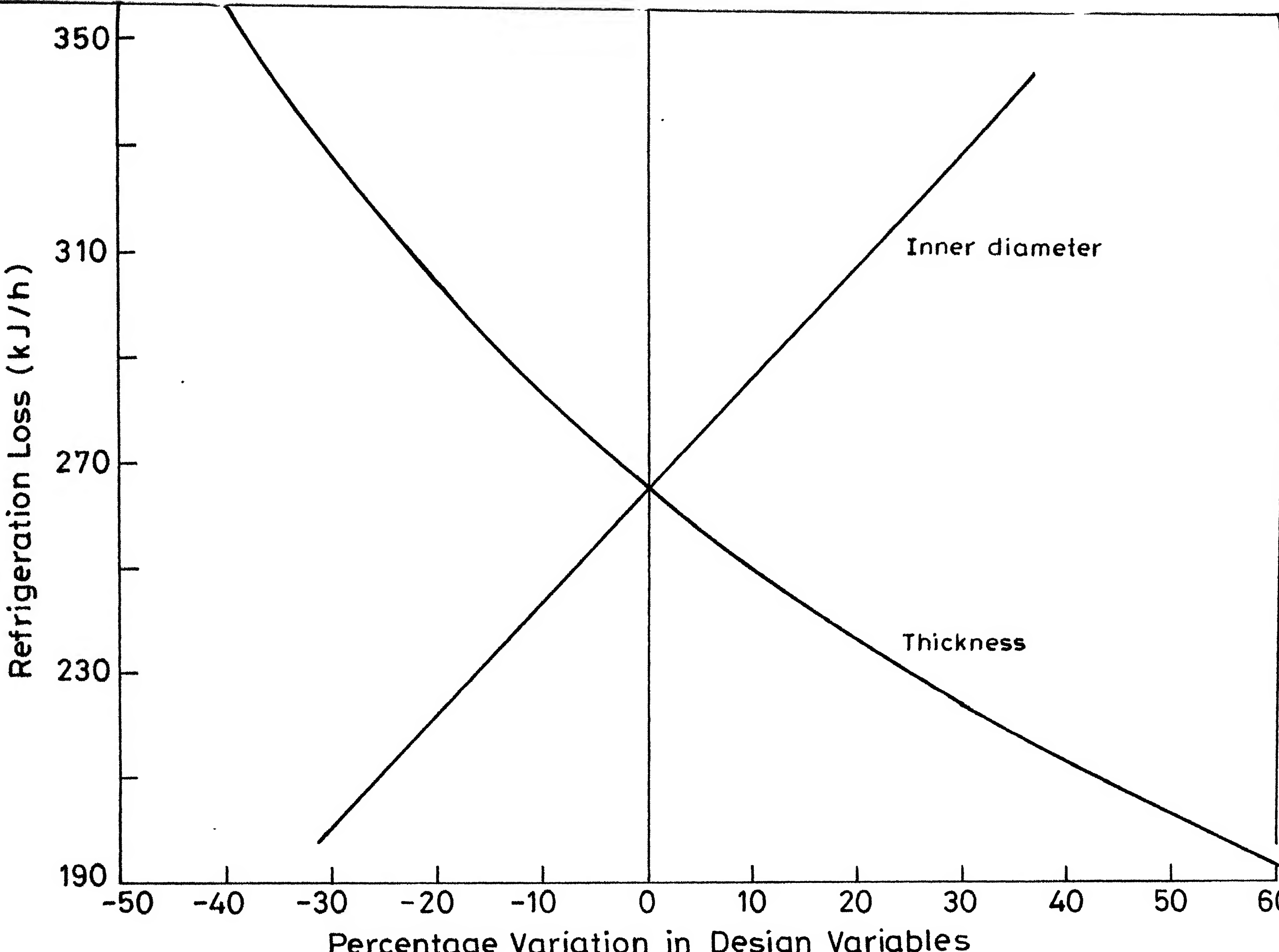

Fig. 4.6 Sensitivity analysis of refrigeration loss.

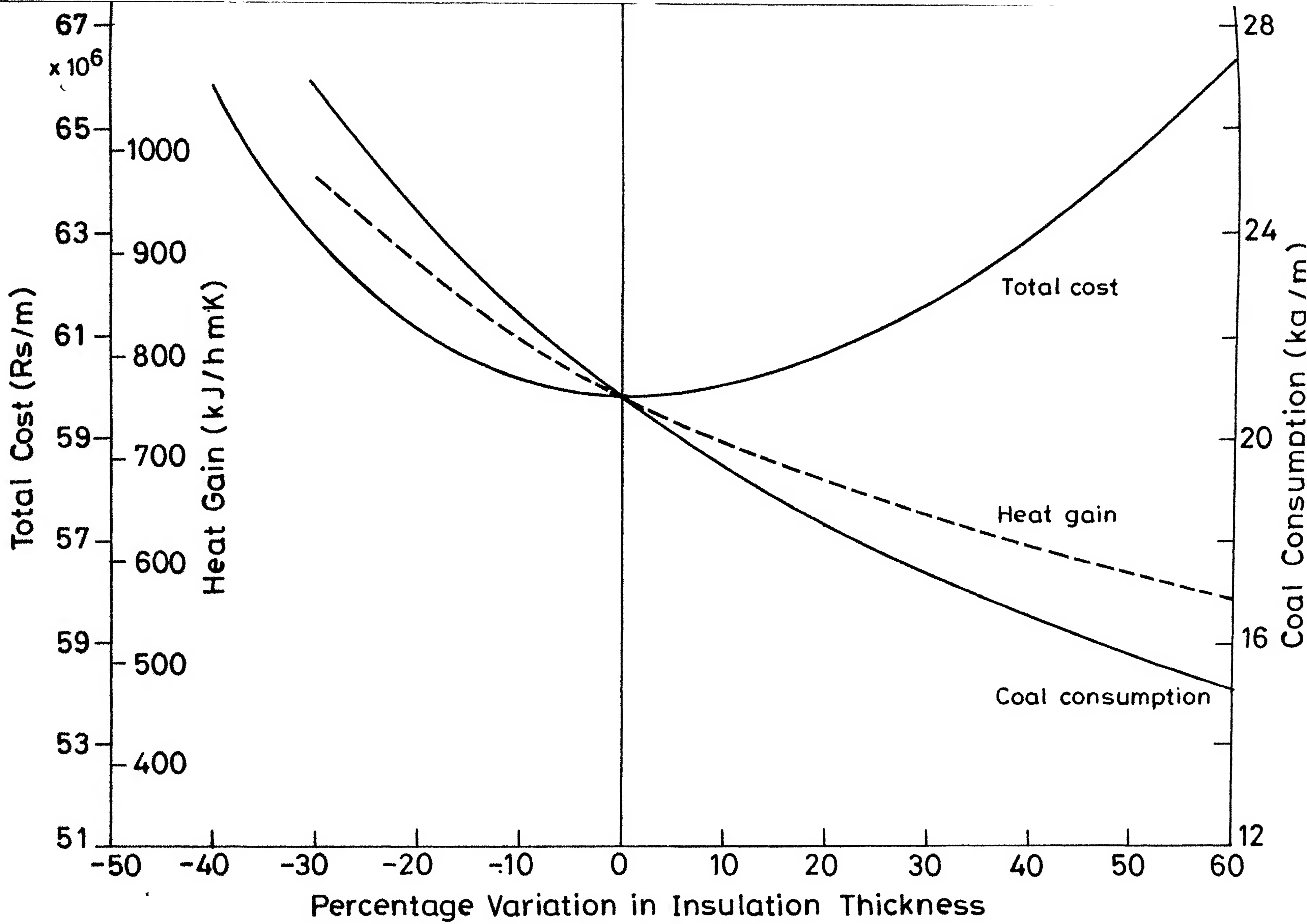

Fig.4.7 Percentage variation in total cost, heating loss and coal consumption with insulation thickness.

CHAPTER-5

CONCLUSIONS AND SUGGESTIONS

5.1 CONCLUSIONS

From the present study the following conclusions are arrived at:

1. The total cost and the electrical energy are found to be more sensitive to the inner diameter and less sensitive to the insulation thickness for the pipe and duct systems.
2. Optimum velocity of water in the pipe system is found to lie in the range of 1.15 to 1.53 m/s for volume flow rate from 1 to 25 m^3/min and ambient temperature 17 C to 47 C: whereas the optimum velocity of the air in the duct system lies in the range of 8.0 to 10.9 m/s for volume flow rate from 10 to 2000 m^3/min and ambient temperature 32 C to 47 C.
3. From the energy conservation point of view 30% to40% more insulation looks to be a reasonable choice to save 7.8% to 9.7% electricity for the pipe system and 10.6% to 13.4% electricity for the duct system and 16.5% to 20.5% electricity for the heat loss through insulated pipe.
4. In view of the rising cost and greater demand of electricity it is desirable to select the dimensions larger than the

optimum value in order to save more energy.

5. This program gives the user the ability to compare various alternatives when selecting or specifying insulation. The program can be used to compare not only different insulations, but also different coverings, since the surface emissivity can be specified.

6. Insulation coverings with low emissivities, such as of aluminium are found to produce lower surface temperatures and lower refrigeration losses than coverings with high emissivities for the pipe and duct system.

7. The results are produced for the circular ducts. It can be used for the rectangular duct using the corresponding expressions for the rectangular duct.

8. The expressions and methodology developed are of general nature and can be used for any set of ambient conditions and prevailing costs. Also the method can be used for any fluid for which the properties are available.

9. The new method for determination of optimum dimensions include all the possible fixed costs in the system. Ignoring any of these costs would mean lowering of generality of the problem.

10. This program could also be used to determine the overall heat transfer coefficient, surface temperature and heat loss/gain accurately.

11. A new method for the determination of insulation thickness and inner diameter is envisaged with respect to energy conservation. The usefulness of this method is demostrated

through a typical example which reveals an energy saving of 25% for the pipe system and 20% for the duct system by incurring 3.5% additional cost than the optimum value.

5.5 SUGGESTIONS

1. The best system would be the one which is optimal for all ambient temperatures and wind velocity round the year. But the components of the system has to be selected to meet the worst ambient temperature. However, as the ambient conditions varies, the optimal dimensions will change. So to get more realistic results the present work can be extended to include the probabilistic design philosophy.

2. The availability of discrete values of insulation thicknesses and pipe diameters can be directly considered by using integer non-linear programming search techniques.

3. The accoustical considerations can also be incorporated along with thermal considerations in the optimum design of the pipe and duct systems.

4. The optimum design of pipe and duct systems using the concept of return period can also be studied.

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APPENDIX-A

A.1. Cost of G.S. Sheets for ducting

G.S. Sheet (gauge)	Cost (Rs./m^2)
18	190
20	180
22	165
24	145

A.2. Cost of mild steel class-C pipes for Kanpur city.

Pipe diameter (Inches)	Cost (Rs./m)
5	360
6	440
8	690
10	980
12	1290
14	1500
16	1760
18	2045

APPENDIX-B

B.1 Variation of steam conductivity (kJ/m.h.k) with temperature and pressure.

Pressure (bars)	Temperature (K)								
	373	423	473	523	573	673	773	873	973
1	0.089	0.103	0.119	0.137	0.156	0.197	0.243	0.290	0.339
50	-	-	-	-	0.189	0.216	0.259	0.306	0.355
200	-	-	-	-	-	-	0.333	0.363	0.407

B.2 Variation of steam Kinematic viscosity (m^2/s) with temperature and pressure ($x10^{-6}$)

Pressure (bars)	Temperature (K)								
	373	423	473	523	573	673	773	873	973
1	20.02	26.9	34.3	42.7	52.1	73.6	98.6	126.9	159.3
50	-	-	-	-	0.87	1.24	1.65	2.12	2.65
200	-	-	-	-	-	0.17	0.22	0.28	0.34

B.3 Variation of steam density (kg/m^3) with temperature and pressure

Pressure (bar)	Temperature (K)								
	373	423	473	523	573	673	773	873	973
1	0.598	0.527	0.472	0.426	0.389	0.330	0.288	0.256	0.229
50	-	-	-	-	23.6	20.1	17.5	15.5	13.9
200	-	-	-	-	-	161.6	140.8	124.6	111.8

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